EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING TRIANGULAR WAVY TAPE (TWT) AS INSERTS FOR TUBE SIDE FLOW OF LIQUIDS

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CERTIFICATE

This is to certify that the thesis entitled "**EXPERIMENTAL STUDIES ON HEAT TRANSFER AUGMENTATION USING TRIANGULAR WAVY TAPES (TWT) AS INSERTS FOR TUBE SIDE FLOW OF LIQUIDS**" submitted by SAMIR KUMAR SAHU (109CH0457) in partial fulfilment of the requirement for the award of BACHELOR of TECHNOLOGY Degree in Chemical Engineering at the National Institute of Technology, Rourkela is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any degree or diploma.

Date: 3rd May' 2013

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CONTENTS

ABSTRACT

The present work deals with the introduction of Stainless Steel Triangular Wavy Tape (TWT) as inserts as passive augmentation device, in the inner tube side of a Double Pipe Heat Exchanger. The effect of turbulence on the heat transfer and pressure drop was compared with the values of smooth tube. Experimental studies have also been made on the different modified designs of the TWT by drilling hole of different sizes and at different positions and also by providing baffles. The experiment was conducted using a Double Pipe Heat Exchanger consisting of inner pipe of ID 22 mm and OD 25 mm, and an outer pipe of ID 53 mm and OD 61 mm. The experiments were performed for the inner tube flow rate in the range of 300 – 1200 KPH with Reynolds number varying between 6600 – 23788. It was found that the design TWT 5A-3 2D-2 BS-2 (i.e. a Triangular Wavy Tape with 3 nos. of 5 mm holes on ascending side and 2 nos. of 2 mm holes on downstream side of the wave with a baffle spacing of 2 wavelengths) has maximum value of performance evaluation criteria R_1 (2.38 – 2.72) and has a friction factor almost 11 times that of the smooth tube for constant flow rate or the same Reynolds number.

Keywords: Heat Transfer Augmentation, Triangular Wavy Tape (TWT) Inserts, Modified TWT Inserts.

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NOMENCLATURE

A_i	Heat transfer area, m^2
A_{xa}	Cross- section area of tube with twisted tape, m^2
A_{xo}	Cross-section area of tube, m^2
C_p	Specific heat of fluid, J/Kg.K
d_i	ID of inside tube, m
d_o	OD of inside tube, m
f	Fanning friction factor, Dimensionless
f_a	Friction factor for the tube with inserts, Dimensionless
f_o	Theoretical friction factor for smooth tube, Dimensionless
g	acceleration due to gravity, m/s^2
Gz	Graetz Number, Dimensionless
h	Heat transfer coefficient, $W/m^2 °C$
h_a	Heat transfer coefficient for tube with inserts, $W/m^2 °C$
h_o	Heat transfer coefficient for smooth tube, $W/m^2 °C$
h_i(exp)	Experimental Heat transfer coefficient, $W/m^2 °C$
h_i(theo)	Theoretical Heat transfer coefficient, $W/m^2 °C$
L	heat exchanger length, m
LMTD	Log mean temperature difference, °C
m	Mass flow rate, kg/sec
N	Number of Tubes
Nu	Nusselt Number, Dimensionless

P	Pumping Power
Pr	Prandtl number, dimensionless
Q	Heat transfer rate, W
Re	Reynolds Number, Dimensionless
R1	Performance evaluation criteria based on constant flow rate, Dimensionless
R3	Performance evaluation criteria based on constant pumping power, Dimensionless
U_i	Overall heat transfer coefficient based on inside surface area, $W/m^2{}°C$
v	flow velocity, m/s
Δh	Height difference in manometer, m
ΔP	Pressure difference across heat exchanger, N/m^2
ΔT_i	Temperature drop from inner wall of pipe to fluid, ^0C

Greek Symbols

μ	Viscosity of the fluid, N s/m^2
μ_b	Viscosity of fluid at bulk temperature, N s/m^2
μ_w	Viscosity of fluid at wall temperature, N s/m^2
ρ	Density of the fluid, kg/m^3
η	$R_1/(f_a/f_o)^{1/3}$, dimensionless

Chapter 1

INTRODUCTION

INTRODUCTION:

In the past few years, heat transfer technology be it in the form of conduction, convection or radiation has been widely applied to heat exchanger applications in refrigeration, automotive, process industries etc. such as in aviation and spacecraft engineering, power engineering, chemical, petroleum refining and food stuff industries, refrigerating and cryogenic engineering etc.

Both the overall dimensions and mass of the employed heat exchangers are increasing continuously with the unit power and volume of production. Huge amount of alloyed steels and non-ferrous metals are being used for manufacturing heat exchanger.

It's an immediate problem to reduce the overall dimensional characteristics of heat exchangers. The urgency to increase the thermal performance of heat exchangers, thereby effecting energy, material and cost savings have led to development and use of many techniques termed as "Heat Transfer Augmentation" Techniques. Augmentation techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger. These techniques are also referred to as "Heat Transfer Enhancement" or "Intensification".

Moreover, as a heat exchanger becomes older, the resistance to heat transfer increases owing to fouling or scaling. These problems are more common for the heat exchangers used in marine applications and chemical process industries. In some peculiar applications, such as heat exchangers dealing with fluids of low thermal conductivity and desalination plants, there is a need to increase the heat transfer rate. The heat transfer rate can be improved by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an already existing heat exchanger at an economic pumping power, various techniques have been proposed in recent years and are briefly discussed in the following sections.

For experimental work, stainless steel triangular wavy tape and its different modified designs are used and its effect on performance evaluation criteria and pressure drop has been studied.

Chapter 2

LITERATURE REVIEW

2.1. CLASSIFICATION OF AUGMENTATION TECHNIQUES [1,2]:

The broadly accepted classification of enhancement or augmentation techniques divides the techniques into three different categories:

1. Passive Techniques
2. Active Techniques

   The difference between the two is that the latter requires external power to bring about the effect.

3. Compound Techniques

1. **Passive Techniques:** These generally use surface or geometric modifications to the flow channel by incorporating inserts or additional devices. These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. As these techniques do not require any direct input of external power; they thus, hold the advantage over the active techniques. They promote higher heat transfer co-efficient by disturbing or altering the existing flow behaviour (except for extended surfaces). In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Heat Transfer Augmentation by these techniques can be achieved by using the following modifications.
   a. *Treated Surfaces:* These techniques include the fine-scale alteration of the surface finish or application of a coating (continuous or discontinuous). They are generally used for boiling and condensing; the roughness height is below that which affects single-phase heat transfer.
   b. *Rough Surfaces:* These surfaces can be produced in several configurations ranging from random sand-grain roughness to discrete protuberances. The configuration is specifically chosen to disturb the viscous sub-layer rather than to increase the heat transfer area.
   c. *Extended Surfaces:* These surfaces mostly in the form of fins are now regularly employed in many heat exchangers to increase the heat transfer surface area, especially on the side with the highest thermal resistances.
   d. *Displaced Enhancement Devices:* These are the inserts primarily used in confined forced convection. These inserts are inserted into the flow channel so as to

indirectly improve energy transport at the heated surface by displacing the fluid from the surface of the duct with bulk fluid from the core flow.

e. *Swirl Flow Devices:* These consists of a number of geometric arrangements or tube inserts for forced flow that create rotating or secondary flow. Some of the different types are Inlet Vortex Generators, Twisted Tape Inserts, Stationary Propellers and Axial-Core Inserts with a screw type winding. They can be used for both single phase flow and two-phase flows.

f. *Coiled Tubes:* These tubes leads to more compact heat exchangers. The secondary flows or vortices are generated due to curvature of coils that promote higher single phase heat transfer coefficients as well as improvement in most regimes of boiling.

g. *Surface Tension Devices:* These techniques include wicking or grooved surfaces that direct and improve the flow of fluid to boiling surfaces and from condensing surfaces. Many manifestations of devices involving capillary flow is also possible.

h. *Additives for Liquids:* These include solid particles, soluble trace additives and gas bubbles in single phase flows and trace additives which reduce the surface tension of the liquid for boiling systems.

i. *Additives for Gases:* Additives for gases are liquid droplets or solid particles, which are introduced in single phase gas flows either as dilute-phase (gas-solid suspensions) or dense-phase (fluidised beds).

2. **Active techniques:** These techniques require the use of external power to facilitate the desired flow modifications and improvement in the rate of heat transfer. Thus, these techniques are more complex from the use and design point of view. It finds very limited practical applications. As compared to passive techniques, these techniques have not shown much potential as it is very difficult to provide external power input in many cases. Heat Transfer Enhancement by this technique can be achieved by incorporating one of the following methods.

   a. *Mechanical Aids:* In this, the stirring of the fluid is done by mechanical means or by rotating the surface. Another type is surface "Scrapping", which is widely used in the chemical process industry for batch processing of viscous liquids.

   b. *Surface Vibration:* They are applied in single phase flows to obtain higher convective heat transfer coefficients, at either low or high frequency. This is possible only in certain circumstances as the vibrations of sufficient amplitude to affect the heat transfer may destroy the heat exchanger itself.

c. *Fluid Vibration:* This kind of vibration augmentation technique is employed for single phase flows. Instead of applying vibrations to the surface, pulsations are created in the fluid itself. It is the practical type of vibration augmentation because of large mass of most heat exchangers.
d. *Electrostatic Fields:* Electrostatic fields from a AC or DC source can be applied in different ways to dielectric liquids to cause bulk mixing or disruption of flow in the vicinity of heat transfer surface to enhance heat transfer.
e. *Injection:* It is utilized by supplying gas to a stagnant or flowing liquid through a porous heat transfer surface or by injecting similar fluid into the liquid. The surface degassing of liquids can produce augmentation similar to gas injection.
f. *Suction:* This can be used for both single phase and 2-phase heat transfer process. It involves vapour removal, in nucleate or film boiling, or fluid withdrawal, in single phase flow, through a porous heated surface.

3. **COMPOUND TECHNIQUES:** A compound technique is the one which involves the simultaneous combination of two or more of the above techniques with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

2.2. PERFORMANCE EVALUATION CRITERIA [1]:

The performance objectives are defined such that the augmented exchanger is required to do a better 'job' than the reference design for established design constraints. In most of the practical applications of the augmented techniques, the following performance objectives, with established design constraints and conditions, are normally considered for the optimization of the use of a heat exchanger:

1. Increase the heat duty of an existing heat exchanger without altering the pumping power (or pressure drop) or flow rate requirements.
2. Reduce the approach temperature difference between the two heat-exchanging fluid streams for a specified heat load and size of an exchanger.
3. Reduce the size or heat transfer surface area requirements for a specified heat duty and pressure drop or pumping power.

4. Reduce the process stream's pumping power requirements for a given heat load and exchanger surface area.

It may be noted that objective 1 allow for an increase in heat transfer rate, objective 2 and 4 offer reduced operating costs or energy costs, and objective 3 offer material savings and reduction in capital costs.

Different Criteria used for evaluation of performance of the single phase flow are:

Fixed Geometry (FG) Criteria: The area of flow cross section (N and d_i) and tube length L are kept constant. The criterion is generally applicable for retrofitting the smooth tubes of an existing exchanger with enhanced tubes, thereby maintaining the same basic geometry and size (N, d_i, L). The objectives can then be to increase the heat load Q for the same approach temperature ΔT_i and mass flow rate *m* and pumping power P; or decrease ΔT_i or P for fixed Q and *m* or P; or reduce P for fixed Q.

Fixed Number (FN) Criteria: The flow cross sectional area or frontal area (N and d_i) is kept constant, and the heat exchanger length is allowed to vary. Here the objectives are reduction in either the heat transfer area (A → L) or the pumping power P for a fixed heat load.

Variable Geometry (VG) Criteria: The N and L are kept constant, but the tube diameter can be changed. A heat exchanger is generally sized to meet a specified heat duty Q for a fixed process fluid flow rate *m.* Because the tube side velocity reduces in such cases so as to accommodate the higher friction losses in the enhanced surface tubes, it becomes necessary to increase the flow area to maintain constant *m.* This is usually accomplished by the use of a greater number of parallel flow circuits.

Table 2.1 Performance Evaluation Criteria [1]

Case	Geometry	m	P	Q	ΔT_i	Objective
FG-1a	N, L, d_i	X			X	Q↑
FG-1b	N, L, d_i	X		X		ΔT_i↓
FG-2a	N, L, d_i		X		X	Q↑
FG-2b	N, L, d_i		X	X		ΔT_i↓
FG-3	N, L, d_i			X	X	P↓
FN-1	N, d_i		X	X	X	L↓
FN-2	N, d_i	X		X	X	L↓
FN-3	N, d_i	X		X	X	P↓
VG-1	---	X	X	X	X	(NL) ↓
VG-2a	(NL)	X	X		X	Q↑
VG-2b	(NL)	X	X	X		ΔT_i↓
VG-3	(NL)	X		X	X	P↓

Bergles et al [3] suggested a set of eight (R_1-R_8) numbers of performance evaluation criteria as shown in Table 2.2.

Table 2.2. Performance Evaluation Criteria of Bergles et al [3].

		Criterion Number							
		R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8
OBJECTIVE	Increase Heat Transfer	X	X	X					
	Reducing Pumping Power				X				
	Reduce Exchange Size					X	X	X	X
FIXED	Basic geometry	X	X	X	X				
	Flow Rate	X						X	X
	Pressure Drop		X				X		X
	Pumping Power			X		X			
	Heat Duty				X	X	X	X	X

It may be noted that FG-1a & FG-2a are similar to R_1 and R_3 respectively.

2.3. SOME OF THE IMPORTANT INVESTIGATIONS USING DIFFERENT TYPES OF TUBE INSERTS:

The tube insert devices deal with the twisted tape inserts, extended surface inserts, wire coil inserts, mesh inserts etc. Twisted tape inserts cause the spiralling of the flow along the tube length. They usually do not have a good thermal contact with the wall of the tube. Wire coil inserts contain a helical coiled spring which acts as a non-integral roughness. Some of the inserts act as an extended surface and help in reduction of the hydraulic diameter. The selection the tube-inserts depend mainly on two factors: costs and performance. The performance comparison for the different types of the tube-inserts is a very useful complement to the usual retrofit design of both the heat exchangers and the networks of heat exchanger. Zimparov et al. [4] discussed about the selection of the tube inserts and compared the thermal and hydraulic performance for the two widely applied tube inserts: wire coil insert and twisted tape insert. From the obtained results, they stressed that the implementation of wire coil inserts meets the requirement of increased heat rate $Q > 1$, whereas use of twisted tape inserts, in most cases, the objective $Q > 1$ cannot be reached. They concluded that the wire coil insert is more attractive heat transfer technique compared to the twisted tape insert, when a retrofit design has to be used. This may be probably because of the fact that in case of turbulent flow the objective is to disturb the sub-viscous layer which the wire coil inserts do. While the twisted tape inserts help to disturb the bulk liquid hence more useful when used in case of laminar flow.

Smith Eiamsa et al. [5] performed an experimental study on turbulent heat transfer and friction flow characteristics in a circular tube equipped with two types of twisted tapes: 1. Typical twisted tapes and 2. Alternate clockwise and counter clockwise twisted tapes (C-CC twisted tapes) as shown in fig. 2.1. Nine different C-CC twisted tapes with three twist ratios, y/W=3.0, 4.0, 5.0, each with three twist angles, $\theta = 30^0$, 60^0, and 90^0 were tested over a range of Reynolds Number 3000-27,000 using water as working fluid under uniform heat flux conditions. The results show that the C-CC twisted tapes provide higher heat transfer rate, performance evaluation factor (R_1) and friction factor than the normal twisted tapes at similar operating conditions and they also reveal that the heat transfer rate of the C-CC tapes increases with the decrease of twist ratio and the increase of the twist angles. Depending upon the above mentioned factors, the mean Nusslet numbers in the tube with C-CC twisted tape inserts are higher than that of the typical ones and the plain tube by around 12.8- 41.9 % and 27.3- 90.5 % respectively. The very small increase in Nusselt numbers may because of turbulent flow (Re = 3000 – 27,000) .

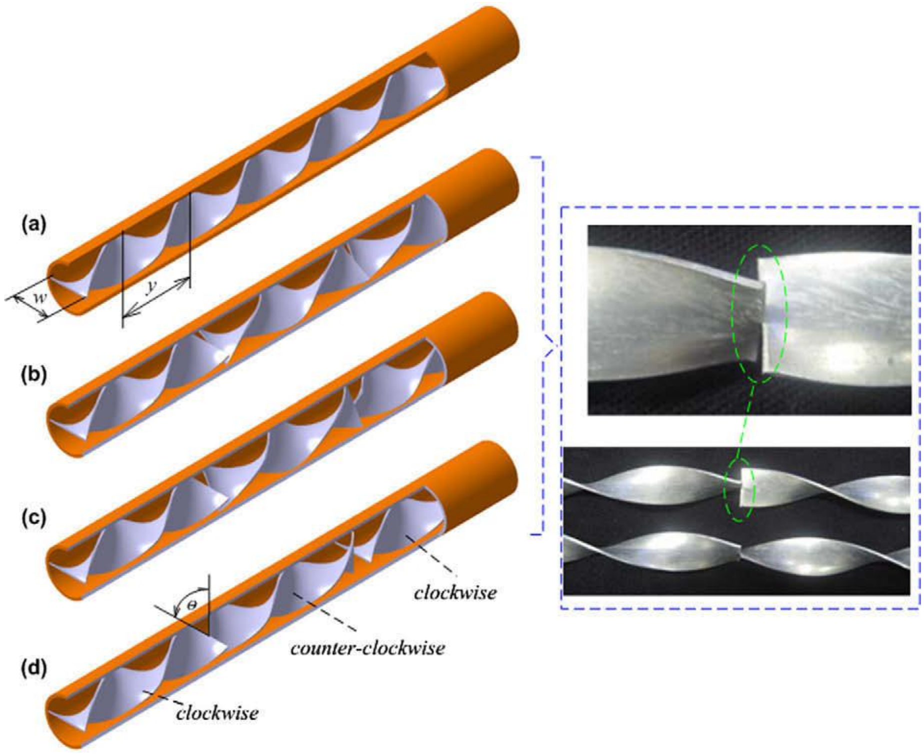


Fig. 2.1. : Test tube fitted with twisted tapes: (a) The typical twisted tape (b) C-CC twisted tape with θ=30$^{\theta}$ (c) C-CC twisted tape with θ=60$^{\theta}$ (d) C-CC twisted tape with θ=90$^{\theta}$.[5]

S. Eiamsa et al [6] reported on heat transfer enhancement using helically twisted tapes (HTTs). The fabrication of each of the helically twisted tapes was done by twisting a straight tape to form a general twisted tape and then bending the twisted tape into a helical structure (fig. 2.2). The experiments were done using HTTs with three twist ratios (y/W) of 2, 2.5 and 3, three helical pitch ratio's (p/D) of 1, 1.5 and 2 for Reynolds Number ranging from 6000 to 20,000. The Conventional helical tape (CHT) was also tested for comparison. The experimental results reveal that HTTs give lower Nusselt Number and friction factor but higher thermal performance ratio (η) than the CHT's at similar conditions of twists ratios and helical pitch ratios. The friction factor and the heat transfer rate increase as the tape twist ratio and helical pitch ratio decrease, while the thermal performance decreases. In the present range, the highest thermal performance ratio achieved by using the largest twist ratio (y/W=3) and helical pitch ratio (p/D=2) at Re=6000 is 1.29.

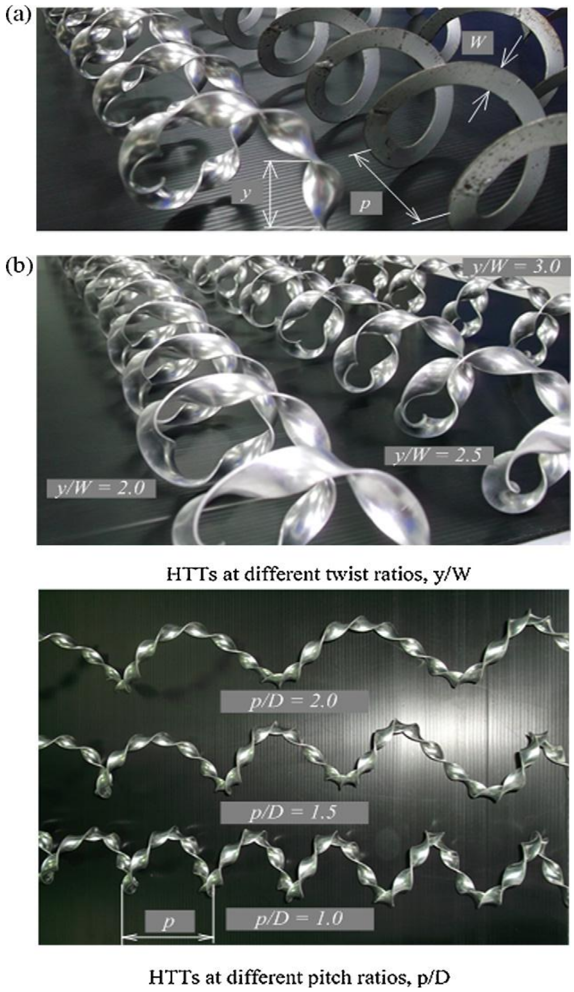


Fig. 2.2. : photographs of helically twisted tapes (HTT) and conventional helical tapes (CHT): (a) geometries of HTT and CHT (b) geometric details of helically twisted tapes, [6]

Smith et al. [7] studied the heat transfer, enhancement efficiency (ratio of the heat transfer coefficient for the inserted tube to that for the plain tube) and friction loss behaviours in a heat exchanger tube equipped with propeller type swirl generators (fig. 2.3) at several pitch ratios (PR) for the Reynolds number ranging from 4,000 to 21,000 under a uniform heat flux condition. The experiments were also performed for different blade numbers of the propeller (N=4, 6 and 8 blades) and for different blade angles ($\theta = 30^0$, 45^0 and 60^0). The results obtained reveal that the tube with propeller inserts provides considerable improvement of the heat transfer rate from the plain tubes by 2.07 to 2.18 times for PR=5, $\theta=60^0$ and N=8. The use of propellers increases the maximum enhancement efficiency by 1.2 times and the friction factor also increases by 3-18 times from that of plain tubes. Therefore, due to strong swirl or rotating flow, the propellers and their blade numbers influence the heat transfer enhancement.

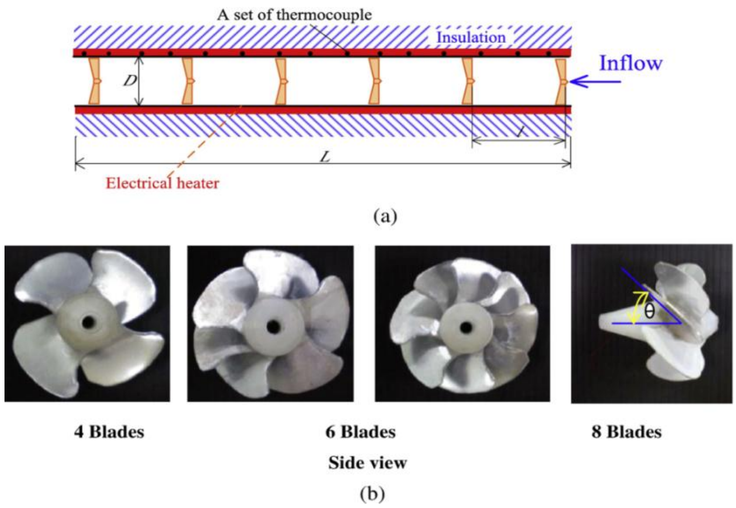


Fig. 2.3. : Test tube with (a) location of thermocouples and propeller types (b) Various propellers types [7]

C. Thianpong et al. [8] experimented on dimpled tube fitted with a twisted tape (fig. 2.4) using air as working medium to study the friction and compound heat transfer behaviours and the impacts of pitch and twist ratio on the average heat transfer coefficient and the pressure loss are determined in a circular tube for a range of Re 12,000-44,000. The Experiments were conducted using plain tube, dimpled tube acting alone and two dimpled tubes with different pitch ratios of dimpled surfaces (PR= 0.7 & 1.0) and three twisted tapes with three different twist ratios (y/W= 3,5 & 7). The Obtained results indicate that friction factor and heat transfer coefficient in the dimpled tube with twisted tape , are higher than those in dimpled tube acting alone and the plain tube and was also found that as the pitch ratio (PR) and twist ratio(y/W) decrease, the friction factor and the heat transfer coefficient increase in the combined devices.

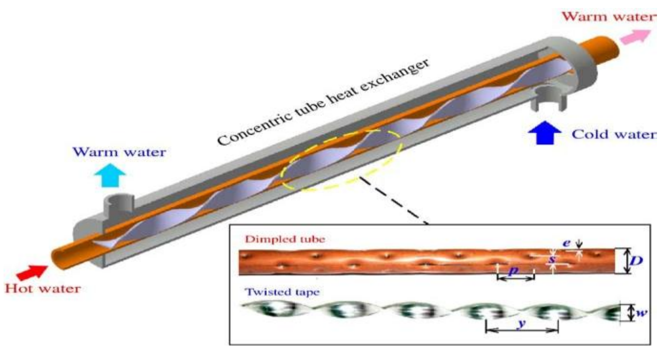
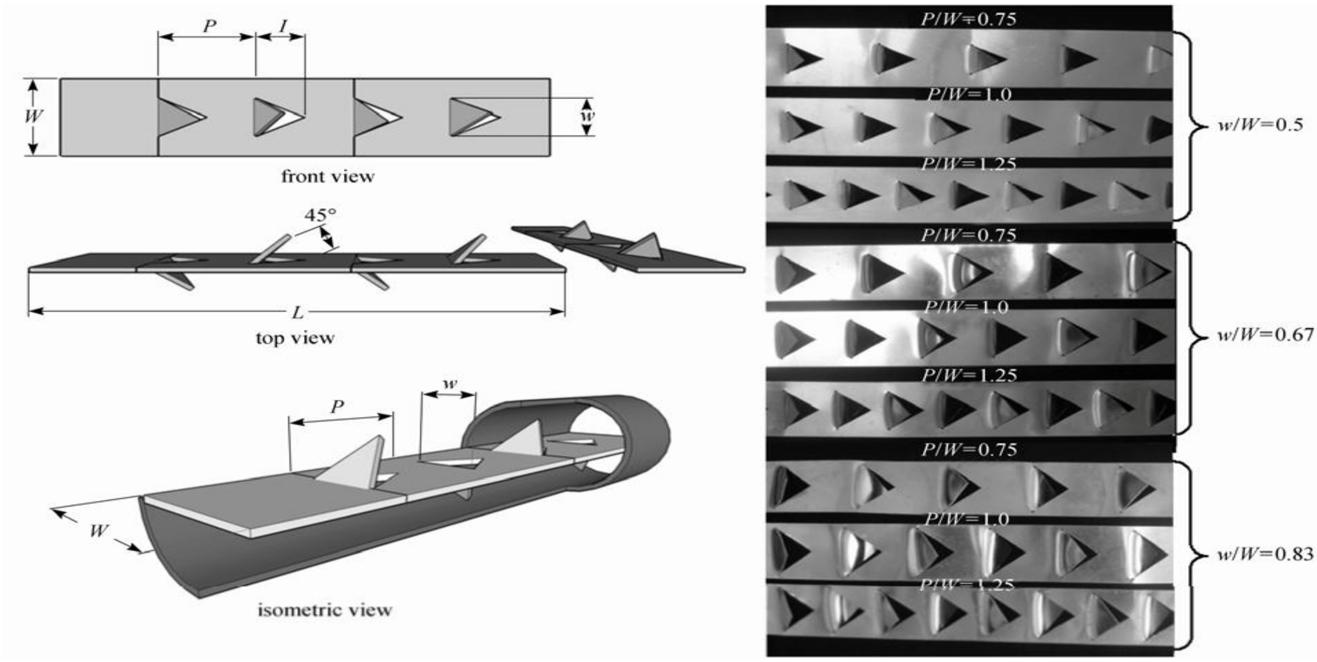
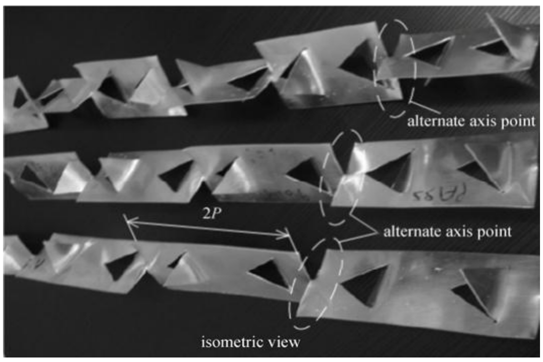


Fig. 2.4. : Dimpled tube and twisted tape [8]

S.Eiamsa et al [9] reported experimental findings of the convective heat transfer and friction behaviours of turbulent tube flow for a straight tape with double-sided delta wings (T-W) (fig. 2.5). In this work, The TW formed on the plate was used as a vortex generator for increasing the heat transfer coefficient by breakdown of thermal boundary layer and by mixing of fluid flow in tubes. The results indicate that for the T-W, the mean Nusselt Number and the friction factor increase by 165% and 14.8 times that of plain tube and the maximum thermal performance factor [ŋ] is 1.19. The investigation also reveals that the friction factor and the heat transfer rate of TW-A is higher than that of the TW.



(a) T-W



(b) TW-A

Fig. 2.5. : Geometry of straight tape with different centre double sided delta wing arrangements [9]

Chapter 3

PRESENT EXPERIMENTAL WORK

3.1. SPECIFICATIONS OF THE HEAT EXCHANGER:

The experiments were carried out in a Double Pipe Heat Exchanger with the following specifications:

Inner pipe ID = 22 mm

Inner pipe OD = 25 mm

Outer pipe ID = 53 mm

Outer pipe OD = 61 mm

Material of Construction = Copper

Heat Transfer Length = 2.43 m

Pressure tapping to pressure tapping length = 2.825 m

Water at room temperature was allowed to flow through the inner pipe while hot water(set point 60^0C) through the annular side in the counter current direction.

3.2. TYPES OF INSERTS USED:

For the present experimental work, Stainless Steel Triangular Wavy Tape (TWT) inserts with thickness of 1 mm and width of 12mm with the wave specifications having wavelength of 3.8 cm (Approx.) and 2 cm of ascending and descending length of wave were used. (Fig. 3.1)

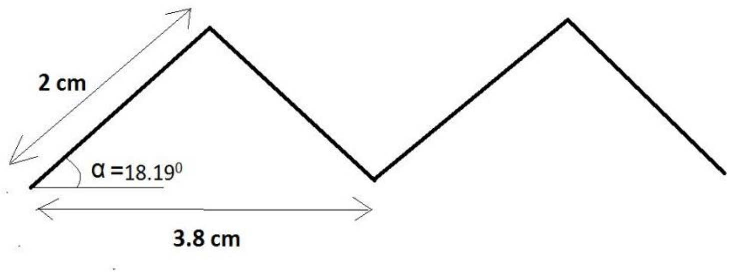


Fig. 3.1 Schematic diagram of the Triangular Wavy Tape (TWT)

Experiments were also conducted with 10 different modifications to the wavy tape. The naming of the different modifications was done with reference to 3 consecutive waves. The different designs along with their naming are as follows.

1. SS Wavy Tape with a 2 mm hole on ascending part of every 4^{th} wave – (TWT 2A-1)
2. SS Wavy Tape with a 2 mm hole on ascending part of two consecutive waves followed by a gap of 1-wave – (TWT 2A-2)
3. SS Wavy Tape with a 2 mm hole on ascending part of each wave – (TWT 2A-3)

4. SS Wavy Tape with a 2 mm hole on ascending part of each wave and on the descending part of every 4th wave – (TWT 2A-3 2D-1)
5. SS Wavy Tape with a 2 mm hole on ascending part of each wave and on descending part of two consecutive waves – (TWT 2A-3 2D-2)
6. SS Wavy Tape with a 2 mm hole on ascending part of first wave replaced by a 5 mm hole – (TWT 5A-1 2A-2 2D-2)
7. SS Wavy Tape with a 2 mm hole on ascending part of two consecutive waves replaced by 5 mm holes – (TWT 5A-2 2A-1 2D-2)
8. SS Wavy Tape with a 5 mm hole on ascending part of each wave – (TWT 5A-3 2D-2)
9. SS Wavy Tape with the design TWT 5A-3 2D-2 along with 0.8 cm baffles at a distance of 4 wavelengths i.e. 4×3.8 cm = 15.2 cm – (TWT 5A-3 2D-2 BS-4)
10. SS Wavy Tape with the design TWT 5A-3 2D-2 along with 0.8 cm baffles at a distance of 2 wavelengths i.e. 2×3.8 cm = 7.6 cm – (TWT 5A-3 2D-2 BS-2)

The details of the design can be known from the code. For Example, for TWT 5A-3 2D-2 BS-2 indicates a Triangular Wavy Tape (TWT) with 3 holes of 5 mm each on ascending, 2 holes of 2 mm each on descending side of the wave and baffle spacing equal to 2 wavelengths.

The followings are the different views of the wave plate with a distance of 2 cm between two bends and the measure between two consecutive crests or troughs is 3.8 cm.

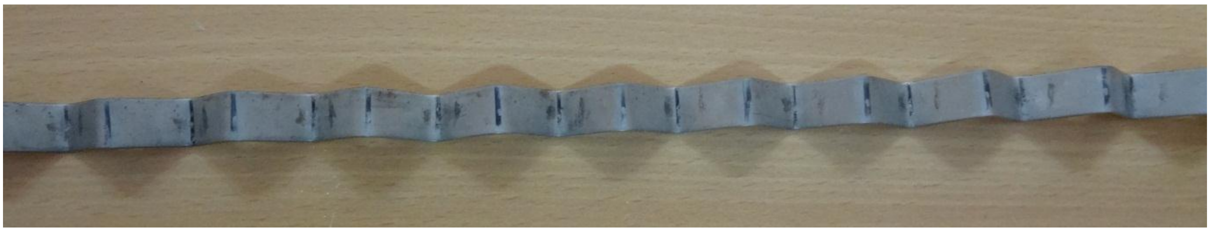


Fig.-3.2a: Top-view of the wavy tape (TWT)

Fig.-3.2b: Isometric-view of the wavy tape (TWT)

Fig. -3.2c : Design model TWT 2A-1

Fig. -3.2d : Design model TWT 2A-2

Fig. -3.2e : Design model TWT 2A-3

Fig. -3.2f : Design model TWT 2A-3 2D-1

Fig. -3.2g : Design model TWT 2A-3 2D-2

Fig. -3.2h : Design model TWT 5A-1 2A-2 2D-2

Fig. -3.2i : Design model TWT 5A-2 2A-1 2D-2

Fig. -3.2j : Design model TWT 5A-3 2D-2

Fig. -3.2k : Design model TWT 5A-3 2D-2 BS-4

Fig. -3.2l : Design model TWT 5A-3 2D-2 BS-2

3.3.FABRICATION OF WAVE PLATES FROM STAINLESS STEEL TAPES:

The stainless steel tapes of width 12mm and length 120 cm were taken and were fitted in a Bench Vice and were hammered slightly on alternate sides at every 2 cm distance so as to obtain a zigzag shape resembling a wavy tape. The wavy tape obtained was irregular in wavelength. It was regularized and the wavelength was made uniform by drawing a triangular wavelength of the specified dimensions on a card-board and then with the help of wrench and plier, the irregular wavy tape was aligned along the drawing. Thus, a triangular wavy tape of wavelength 3.8 cm was obtained.

The holes on the wavy tape were drilled using two drill bits of size 2mm and 5 mm respectively. The markings for the hole were done using a marker and then the markings were slightly hammered so that the wavy tape does not slip away while drilling.

The baffles were made out of the same type of stainless steel tape of length 0.8 cm with the help of hacksaw. The baffles were then attached to the wavy tape with the help of m-seal at intervals of required wavelengths.

3.4.EXPERIMENTAL SET UP:

The Fig.3.2 shows the schematic diagram of the experimental setup. Basically, it is a double pipe heat exchanger consisting of an inner pipe of ID 22 mm and OD 25 mm, and an outer pipe of ID 53 mm and OD 61 mm. The apparatus is also equipped with two rotameters for continuously measuring and maintaining the particular flow rate. One for measuring hot water flow and another for measuring flow rate of cold water. The source for the cold water was from a bore-well from where water was pumped through a submersible pump. There is another tank of capacity 500 litres which has an in-built heater and pump for providing hot water at a particular desired temperature and flow rate. It is also equipped with a digital temperature indicator connected to four RTD sensors. They have four different sensors situated at different locations to give the temperatures T1-for Inner Tube Inlet, T2- for Inner Tube Outlet, T3-for Outer Tube Inlet and T4-Outer Tube Outlet.

The flow rate of the Hot Water was maintained constant at 1000 kg/hr throughout the experimental procedure. There is a U-tube manometer for measuring the pressure drop in the inner tube. It consists of two limbs well connected with the two points in the inner pipe. The fluid filled inside the manometer is Carbon Tetra-Chloride (CCl_4) with Bromine to give it a pinkish colour.

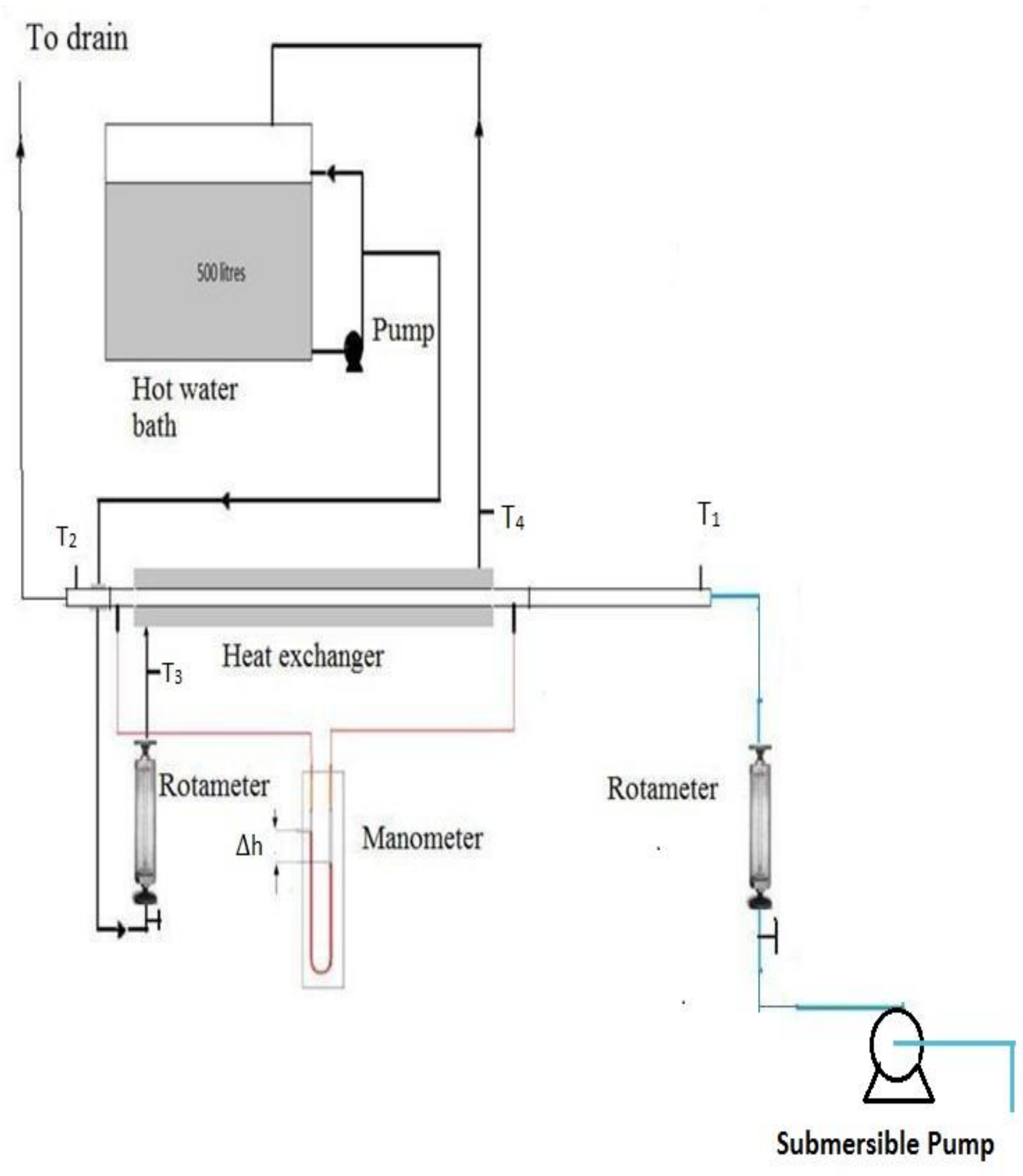


Fig. 3.3. : Schematic diagram for the experimental setup

Fig.-3.4: Photograph of the Experimental Setup.

3.5 EXPERIMENTAL PROCEDURE:

1. All the RTD and the Rotameter for the cold water flow rate were calibrated first.
    i. For the rotameter calibration, we collected water in a bucket, and simultaneously time and weight were noted. Thus mass flow rate was calculated.
    ii. We repeated the same procedure for three times for each particular reading and then average of all the three was taken. The readings are given in A.1.1.
    iii. For RTD calibration, all the four RTDs were simultaneously dipped in the same water bucket and readings were noted. Making T_1 as reference, corrections were made to other RTDs values (i.e. T_2-T_4) accordingly, (App. A.1.2).
2. Standardization of the set-up:

    Before beginning with the experimental study on friction and heat transfer in Heat Exchanger using inserts, the standardization of the experimental setup was done by obtaining the friction factor & heat transfer results for the smooth tube & comparing the obtained data with the standard equations available.
3. For friction factor determination:

    Pressure drop was measured for each flow rate with the help of manometer at room temperature.
    i. The U-tube manometer used carbon tetrachloride as the manometer liquid.
    ii. Air bubbles were removed from the manometer so that the liquid levels in both the limbs are same when the flow rate is made zero. The air bubbles were removed by removing the clips attached to the open ends of the pipes connected to the U-Tube limbs and then allowed the water to flow the open ends in a controlled manner by controlling the flow with the help of hand to ensure that the air bubbles in the manometer escape out. Then, the ends were closed with the help of clips. This procedure was repeated every time the experiment is done.
    iii. Water at the room temperature was allowed to flow through the inner pipe of the Heat Exchanger.
    iv. The manometer reading was then noted.
4. For Heat Transfer Coefficient calculations:
    i. Then, heater was put on to heat the water to 60^0C and maintain the constant temperature of 60^0C in a water tank of capacity 500 litres. The tank is provided with a centrifugal pump and a bypass valve for recirculation of hot water to the tank and to the experimental setup.

ii. Hot water at about 60^{0}C was allowed to flow through the annulus side of Heat Exchanger at 1000 KPH ($\mathbf{m}_h$=0.2778 Kg/sec).

iii. Cold water was simultaneously allowed to pass through the tube side of Heat Exchanger in counter current direction at a desired flow rate.

iv. The water inlet and outlet temperatures for both the inner and outer tube (T_1-T_4) were recorded only after the temperature of both the fluids attained a constant value.

v. The Procedure was repeated for different inner tube flow rates ranging from 0.0601-0.3390 Kg/sec.

5. Preparation of Wilson Chart:

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{d_i}{d_o \times h_o} + \frac{x_w \times d_i}{k_w \times d_l} + R_d$$

Where R_d is the Total dirt resistance - Eq. 3.1

All the resistances, except the first term on the RHS of equation 3.1, are constant for this set of experiments.

For Re>10000, Seider Tate equation for smooth tube is of the form: $h_i = A \times Re^{0.8}$

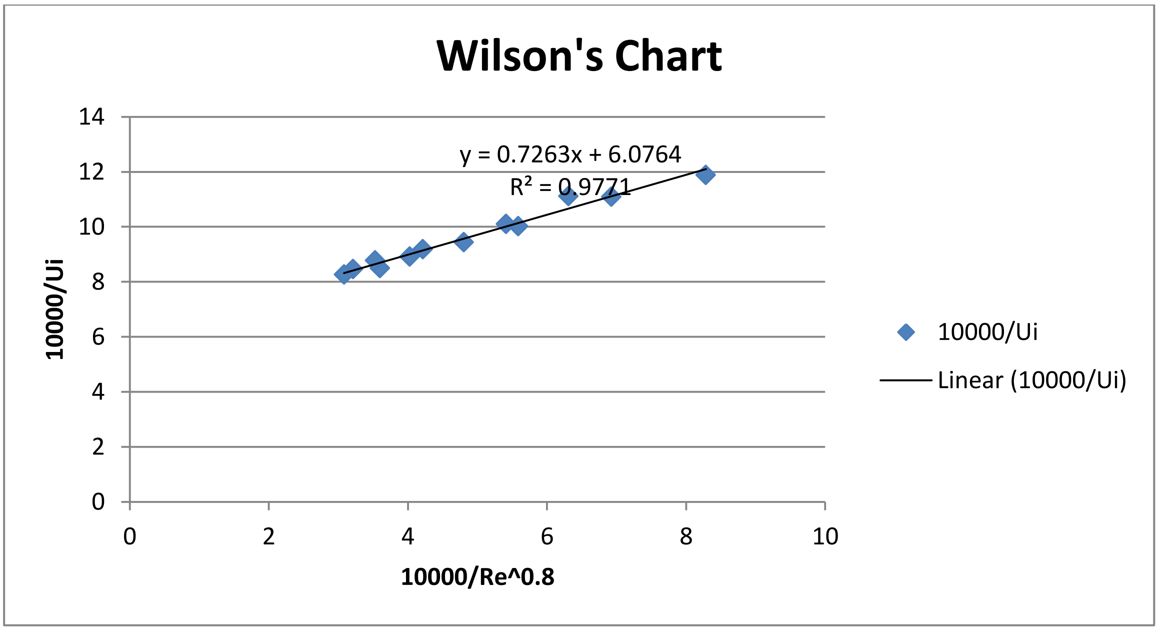


Fig. 3.5 : Plot of Wilson's Chart

Therefore, Eq. (3.1) can be written as

$$\frac{1}{U_i} = \frac{1}{A \times Re^{0.8}} + K$$

Eq. 3.1 a

K is to be found from the Wilson's chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

K= 6.0764 X 10^{-4}

6. After confirmation of the validity of the experimental values of friction factor & heat transfer coefficient in smooth tube with standard equations, friction factor & heat transfer studies with inserts were conducted.
7. The friction factor & heat transfer observations & results for all the cases are presented in Tables A.2.1-A.2.12 & A.3.1-A.3.12 respectively.

3.6 STANDARD EQUATIONS USED: (For Plain Tube)

I. Friction factor (f_o) calculations:

a. For Re < 2100

$$f = \frac{16}{Re}$$

Eq. 3.2

b. For Re > 2100

Colburn's Equation:

$$f = \frac{0.046}{Re^{0.2}}$$

Eq. 3.3

II. Heat Transfer calculations:

i. Laminar Flow:

For Re < 2100

Nu = f(Gz)

Where Gz = $\frac{Re \times Pr \times d_i}{L}$

Eq. 3.4

a. For Gz < 100, Hausen Equation is Used.

$$Nu = 3.66 + \frac{0.085Gz}{1 + 0.045Gz^{0.67}} (\frac{\mu b}{\mu w})^{0.14}$$

Eq. 3.5

b. For Gz > 100, Seider Tate equation is used.

$$Nu = 1.86Gz^{\frac{1}{3}}(\frac{\mu b}{\mu w})^{0.14}$$

Eq. 3.6

ii. Transition Zone:

For 2100 < Re < 10000, Hausen equation is used.

$$Nu = 0.116\left(Re^{2/3} - 125\right) \times Pr^{1/3} \times [\,1 + \left(\frac{D}{l}\right)^{2/3}](\frac{\mu_b}{\mu_w})^{0.14}$$

Eq. 3.7

iii. Turbulent Zone:

For Re > 10000, Dittus-Boelter equation is used.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}$$

Eq. 3.8

Viscosity correction Factor $(\frac{\mu_b}{\mu_w})^{0.14}$ is assumed to be equal to 1 for all calculations as this value for water in present case will be very close to 1 & the data for wall temperatures is not measured.

Chapter 4

SAMPLE CALCULATIONS

4.1 ROTAMETER CALIBRATION:

For 900 Kph (Table No. A.1.1)

Observation No. 1

Weight of water collected = 2.26 Kg

Time = 8.9 sec

$\mathbf{m}_1$ = 0.2539 Kg/sec

Observation No. 2

Weight of water collected = 2.05 Kg

Time = 8.0 sec

$\mathbf{m}_2$ = 0.2563 Kg/sec

Observation No. 3

Weight of water collected = 2.1 Kg

Time = 8.72 sec

$\mathbf{m}_3$ = 0.2408 Kg/sec

$$\mathrm{m} = \frac{\mathrm{m}_1+\mathrm{m}_2+\mathrm{m}_3}{3} = \frac{0.2539+0.2563+0.2408}{3} = 0.2503 \text{ Kg/sec}$$

diff. = 0.11 %

4.2 PRESSURE DROP & FRICTION FACTOR CALCULATIONS: [10]

For Wavy Tape TWT 5A-2 2A-1 2D-2 (Table No. A.2.9)

m= 0.1988 Kg/sec

Experimental friction factor:

$$Area = \frac{\pi \times {d_i}^2}{4} = \frac{\pi \times 0.022^2}{4} = 3.8 \ \times \ 10^{-4} m^2$$

$$v = \frac{m}{A \times \rho_w} = \frac{0.1988}{3.8 \ \times \ 10^{-4} \times \ 1000} = 0.5232 \ m/sec$$

$$\Delta P = \left(\rho_{CCl_4} - \rho_{H_2O}\right) \times g \times \Delta h = (1603 - 1000) \times 9.81 \times 57.1 = 3377.71 N/m^2$$

$$f_a = \frac{\Delta P \times d_i}{2 \times \rho \times L \times v^2} = \frac{3377.71 \times 0.022}{2 \times 1000 \times 2.83 \times 0.5232^2} = 0.048$$

For viscosity calculation:

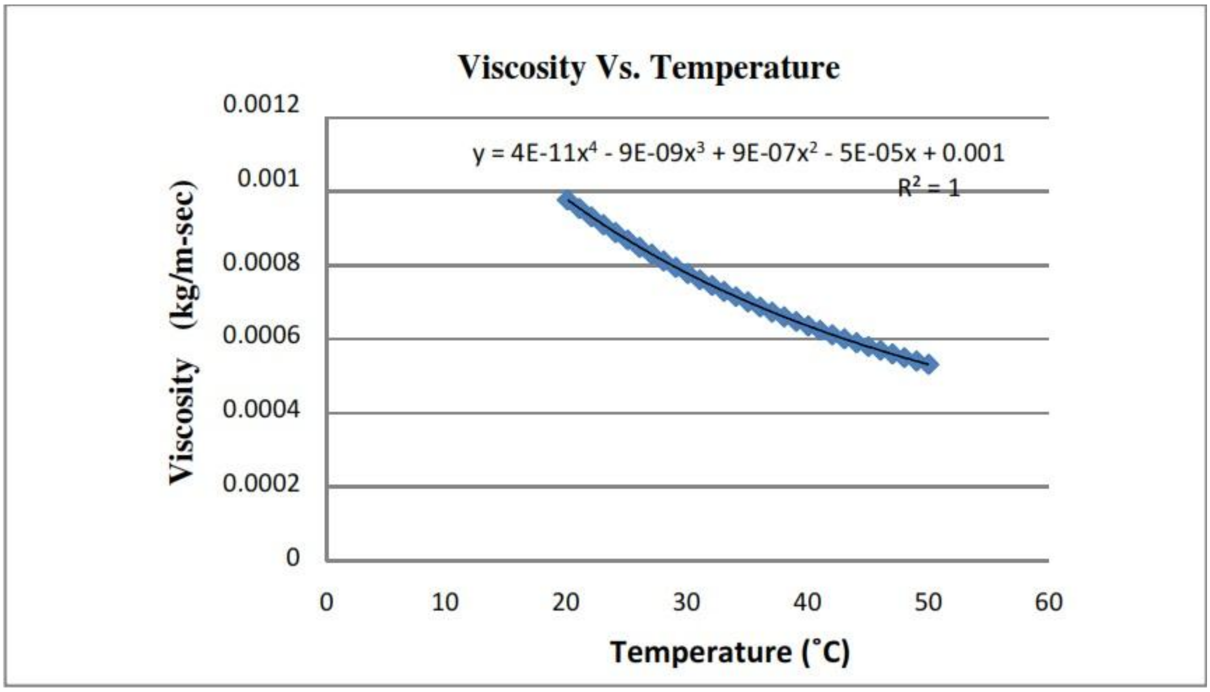


Fig. 4.1: Viscosity vs. Temperature Graph

$$\mu = 4 \times 10^{-11} T^4 - 9 \times 10^{-9} T^3 + 9 \times 10^{-7} T^2 - 5 \times 10^{-5} T + 0.0017$$

Theoretical friction factor calculation for smooth tube:

$$Re = \frac{4 \times m}{\pi \times d_i \times \mu} = \frac{4 \times 0.1988}{\pi \times 0.022 \times 0.00086} = 13384$$

$$f_o = 0.046 \times Re^{-0.2} = 0.046 \times 13384^{-0.2} = 0.0069$$

$$\frac{f_a}{f_o} = \frac{0.048}{0.0069} = 6.96$$

4.3 HEAT TRANSFER COEFFICIENT CALCULATION:

For Wavy Tape 5A-2 2A-1 2D-2 (Table No. A.2.9)

m_c= 0.1988 Kg/sec (700 Kph) & m_h= 0.2778 Kg/sec

NOTE: Temperature correction has already been taken into account while giving data in appendix.

T_1= 26.7^{0}C

T_2= 32.2^{0}C

T_3= 52.6^{0}C

T_4= 48.9^{0}C

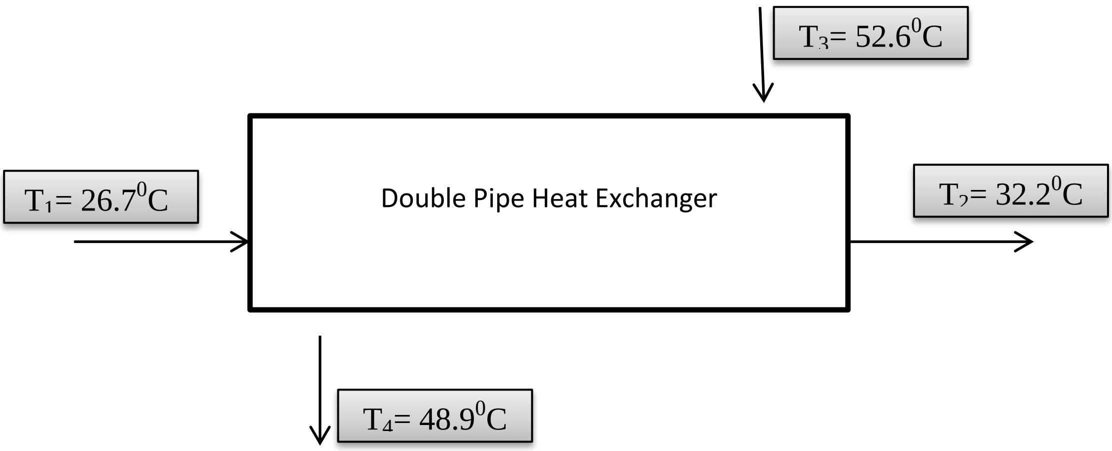


Fig. 4.2: Temperature in different RTDs

$\Delta T_1 = T_4 - T_1$ = (48.9-26.7) = 22.2^{0}C

$\Delta T_2 = T_3 - T_2$ = (52.6-32.2) = 20.4^{0}C

$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}} = \frac{22.2-20.4}{\ln\frac{22.2}{20.4}}$ = 21.29^{0}C

$Q_1 = m_c \times C_{pc} \times (T_2 - T_1)$ = 0.1988×4187×(32.2-26.7) = 4578 W

$Q_1 = m_h \times C_{pc} \times (T_3 - T_4)$ = 0.2778×4187×(52.6-48.9) = 4304 W

Heat balance error = $\frac{4578-4304}{4304} \times 100 = 6.37\%$

$Q_{avg} = \frac{Q_1+Q_2}{2} = 4441\ W$

Heat Transfer Area = $\pi \times d_i \times l = \pi \times 0.022 \times 2.43 = 0.168\ m^2$

$U_i = \frac{Q_{avg}}{A_i \times LMTD} = \frac{4441}{0.1680 \times 21.29} = 1242$ W/m^2 ^{o}C

$$Re = \frac{4 \times m}{\pi \times d_i \times \mu} = \frac{4 \times 0.1988}{\pi \times 0.022 \times 0.00081} = 14210$$

h_i can be calculated using Eq. (3.1.a)

$$\frac{1}{U_i} = \frac{1}{h_i} + K$$

K is found from the Wilson Chart ($1/U_i$ vs. $1/Re^{0.8}$) as the intercept on the y-axis.

K= 6.0764×10^{-4} (Refer fig. 3.4)

$$\frac{1}{h_i} = \frac{1}{U_i} - K = \frac{1}{1242} - 6.0764 \times 10^{-4}$$

$\Rightarrow h_i$ = 5063 W/m^2 ^{o}C $\equiv h_a$.

Theoretical Calculation for smooth tube

$$Nu = 0.023 \times Re^{0.8} \times Pr^{1/3}$$

$$\frac{h_i \times d_i}{k} = 0.023 \times Re^{0.8} \times Pr^{1/3}$$

$$h_i = \frac{0.023 \times k}{d_i} \times Re^{0.8} \times Pr^{1/3}$$

For Prandtl Number calculation:

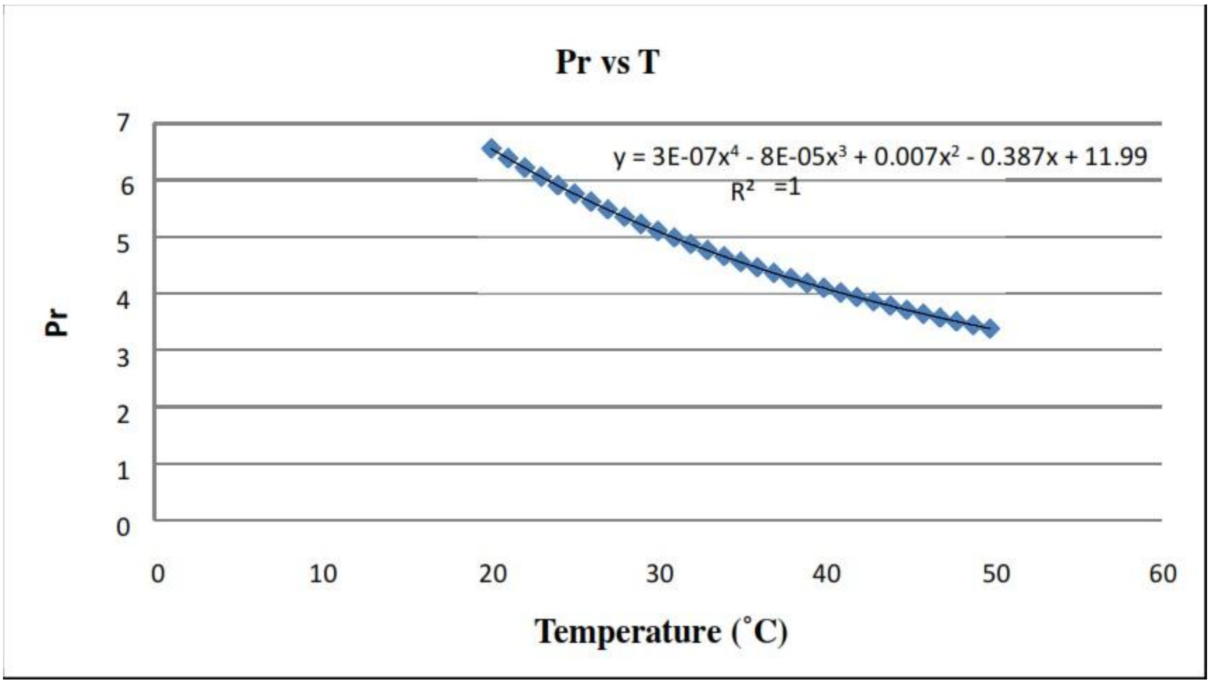


Fig. 4.3 : Prandtl Number Vs. Temperature Graph

$Pr = 3 \times 10^7 T^4 - 8 \times 10^5 T^3 + 0.0072 \times T^2 - 0.3873 \times T + 11.995$

$T_{avg} = \frac{T_1 + T_2}{2} = 29.45\ ^0$C

Pr (at T = T_{avg}) = 5.02

h_i for smooth tube = $\frac{0.023 \times 0.6322}{0.022} \times 14210^{0.8} \times 5.02^{1/3}$

h_i= 2646 W/m^2 ^{o}C ≡ h_o

$R_1 = \frac{h_a}{h_o} = 1.91$

Ŋ = $R_1/(f_a/f_o)^{1/3}$ = 1.91/ $(6.96)^{1/3}$= 1.01

Chapter 5

RESULTS & DISCUSSION

5.1 FRICTION FACTOR RESULTS: [10]

All friction factor results and f_a/f_o values of all the cases are given in the tables A.2.1-A.2.12 [10]. For smooth tube, in almost all range of Reynolds number (neglecting low values of Reynolds number) for the standardization of the setup, the difference of f_{exp} and f_{theo} is very much within the range of $\pm$10% (fig. 5.1). This validated the measurement system of the experiment setup used.

As the ΔH values were very small (0.1-0.8 cm) for low Re & the manometer's least count was 0.1 cm, so we cannot measure those low pressure drops with higher accuracy.

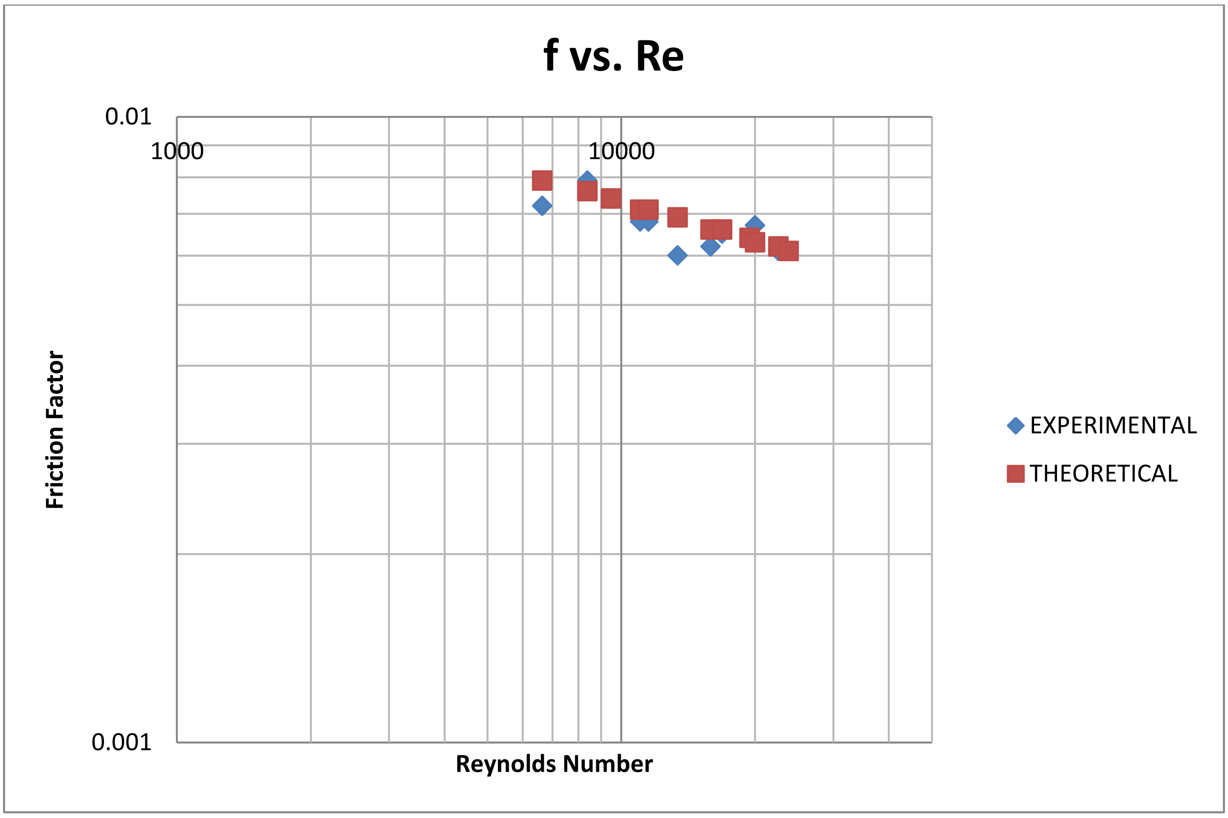


Fig. 5.1 Friction Factor vs. Reynolds Number for smooth tube.

Fig. 5.2 represents the variation of friction factor with Reynolds Number for smooth tube, plain wavy insert and its 10 different modifications with and without baffles. It can be seen from the graph that with every modification there is a slight increase in friction factor and it has been found that the highest friction factor is for the design with baffles having baffle spacing of 2 wavelengths. The inserts with baffles are giving higher friction factor than inserts without baffles because of increase in the degree of turbulence.

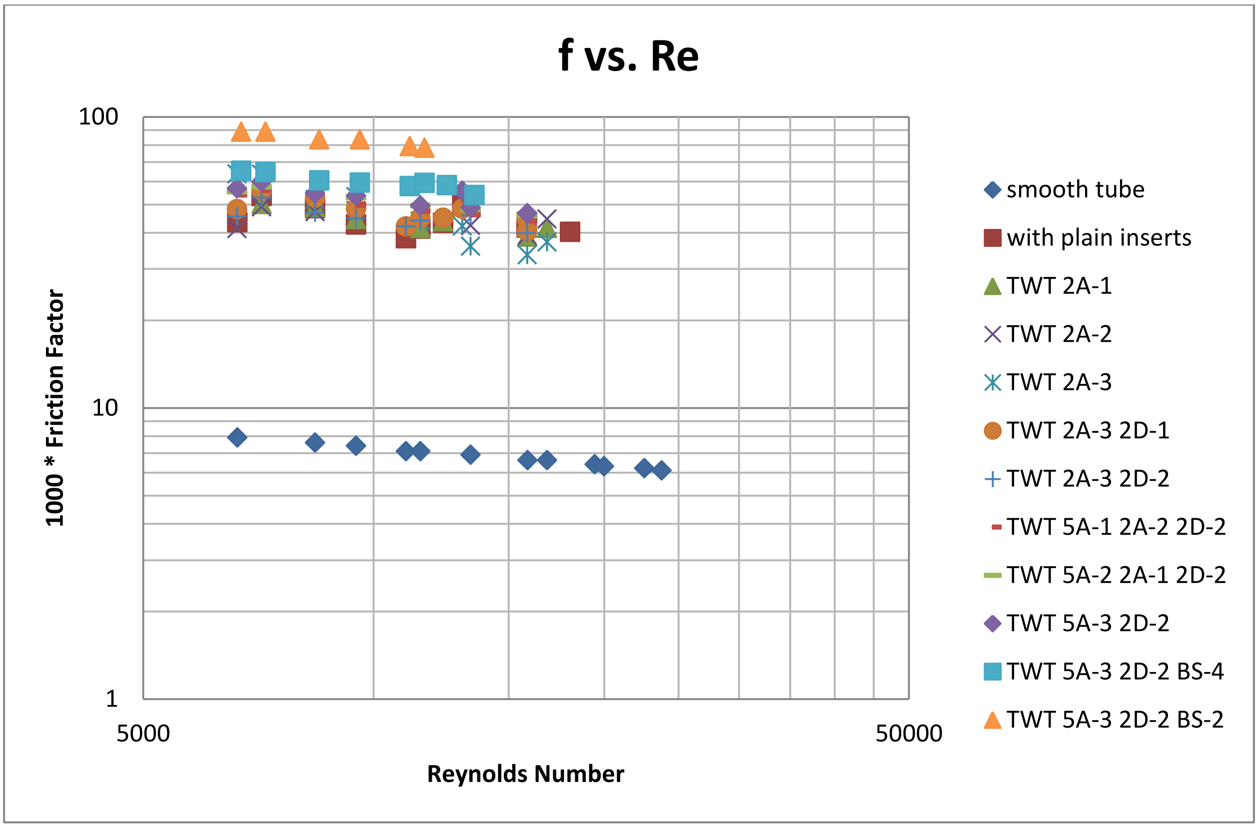


Fig. 5.2 Friction factor vs. Reynolds Number for smooth tube, plain inserts and different modifications to the plain wavy insert.

Fig. 5.3 shows the variation of fa/fo with Reynolds Number for the plain wavy insert and its different modified designs with holes and with baffles. It can be seen that the ratio of fa/fo is highest for the design with baffle spacing of 2 wavelegths.

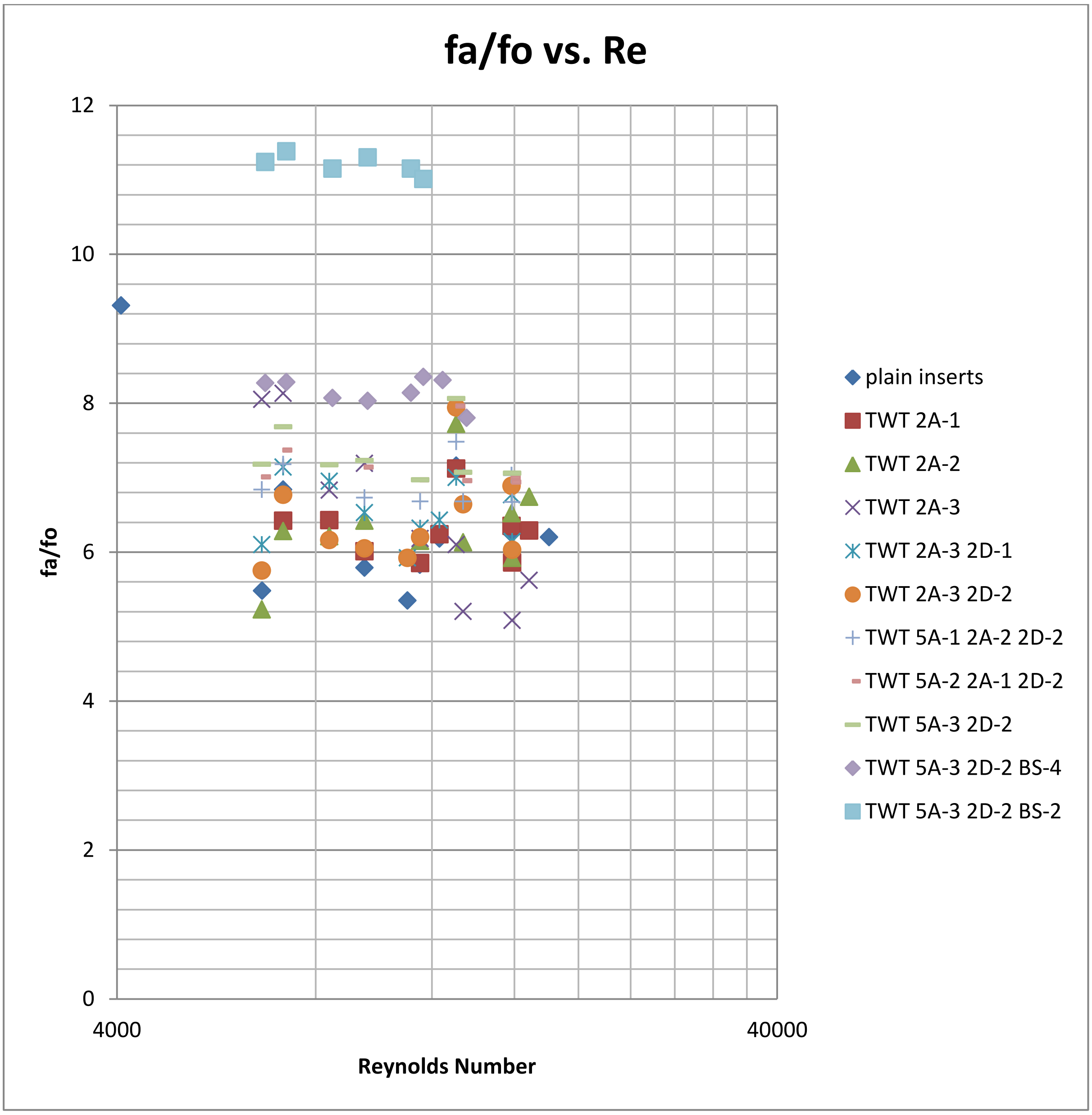


Fig. 5.3 fa/fo vs. Re for plain wavy inserts and its 10 different modified designs.

5.2 HEAT TRANSFER COEFFICIENT RESULTS:

Table A.3.1-A.3.12 gives the heat transfer results for smooth tube, plain wavy insert and its 10 different modified designs along with the corresponding performance evaluation criteria R1 for each of the readings. As can be seen in fig. 5.4, the difference between h_i (exp) & h_i (theo) for smooth tubes is within 13%. So it validates our experimental setup for heat transfer measurements. We have neglected the higher deviation of h_i (exp) and h_i (theo) for low Reynolds number because this can be attributed to the phenomena of natural convection taking place along with the forced convection that is negligible in comparison to forced convection at high Reynolds Number.

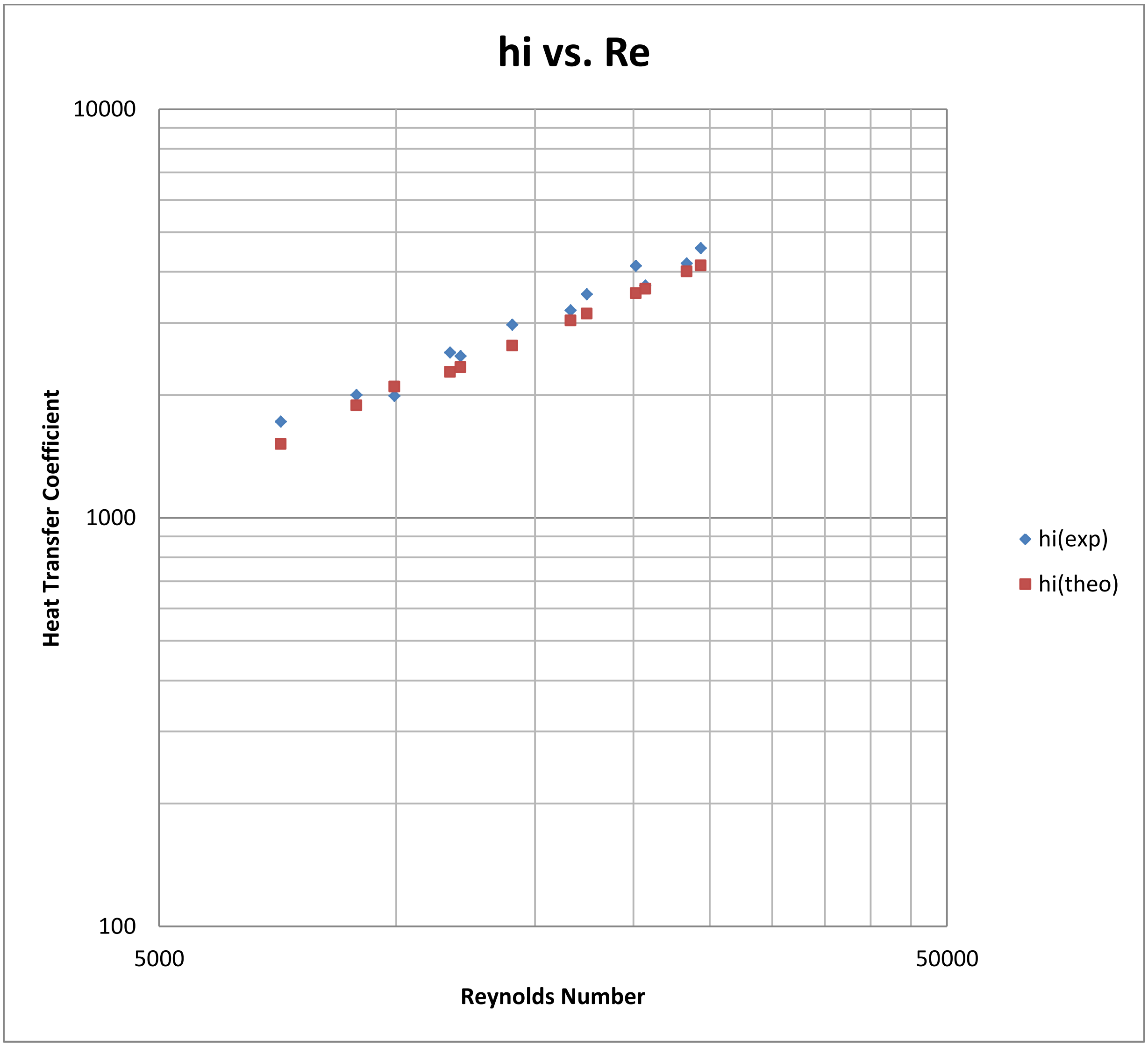


Fig. 5.4 Heat Transfer Coefficient vs. Reynolds Number for smooth tube

Fig. 5.5 represents the variation of heat transfer coefficient vs. Reynolds Number for smooth tube, plain wavy insert and its 10 modified designs with holes and with baffles. As the baffle spacing decreases or the number of holes increase, a higher degree of turbulence is created and hence the heat transfer coefficient increases.

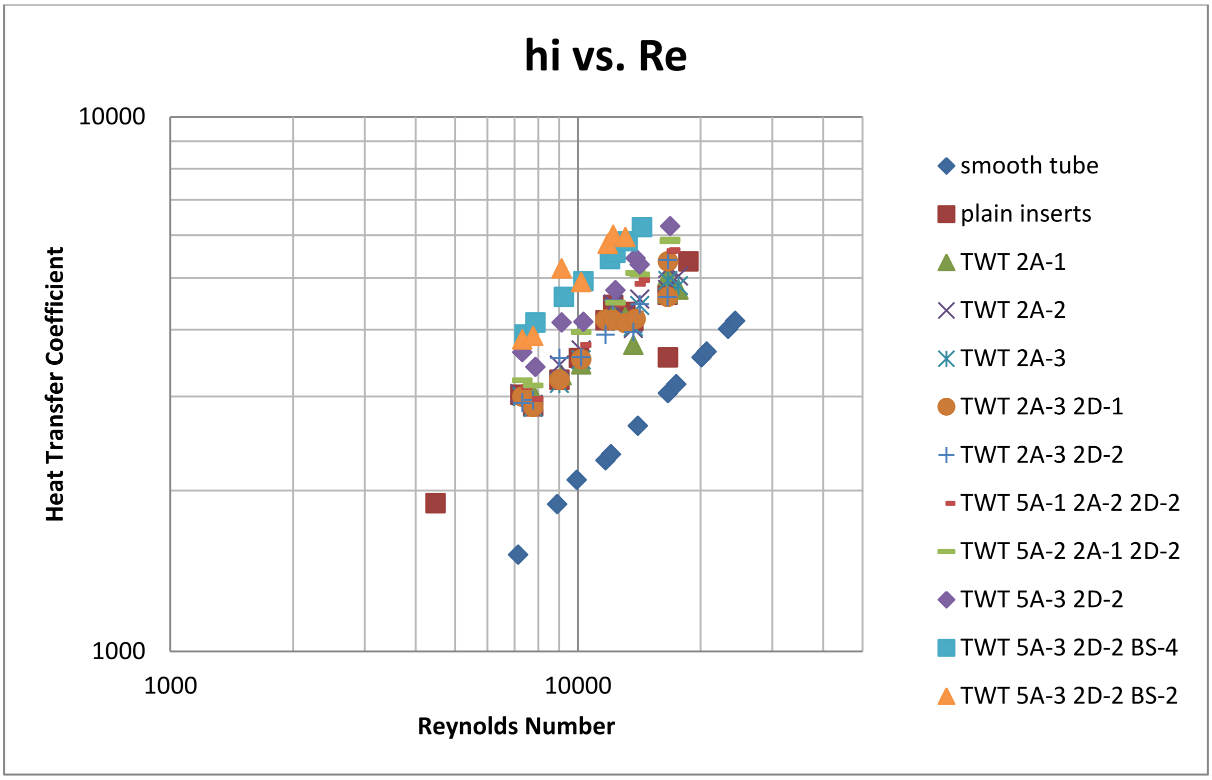


Fig. 5.5 Heat Transfer Coefficient vs. Reynolds Number for smooth tube, plain wavy insert and its 10 modified designs.

Fig. 5.6 shows the plot of performance evaluation criteria R1 vs. Reynolds Number. The Highest value of the R1 is observed in case of design TWT 5A-3 2D-2 BS-2. From this we can conclude that this is the best design out of all the modified designs tested for this experiment.

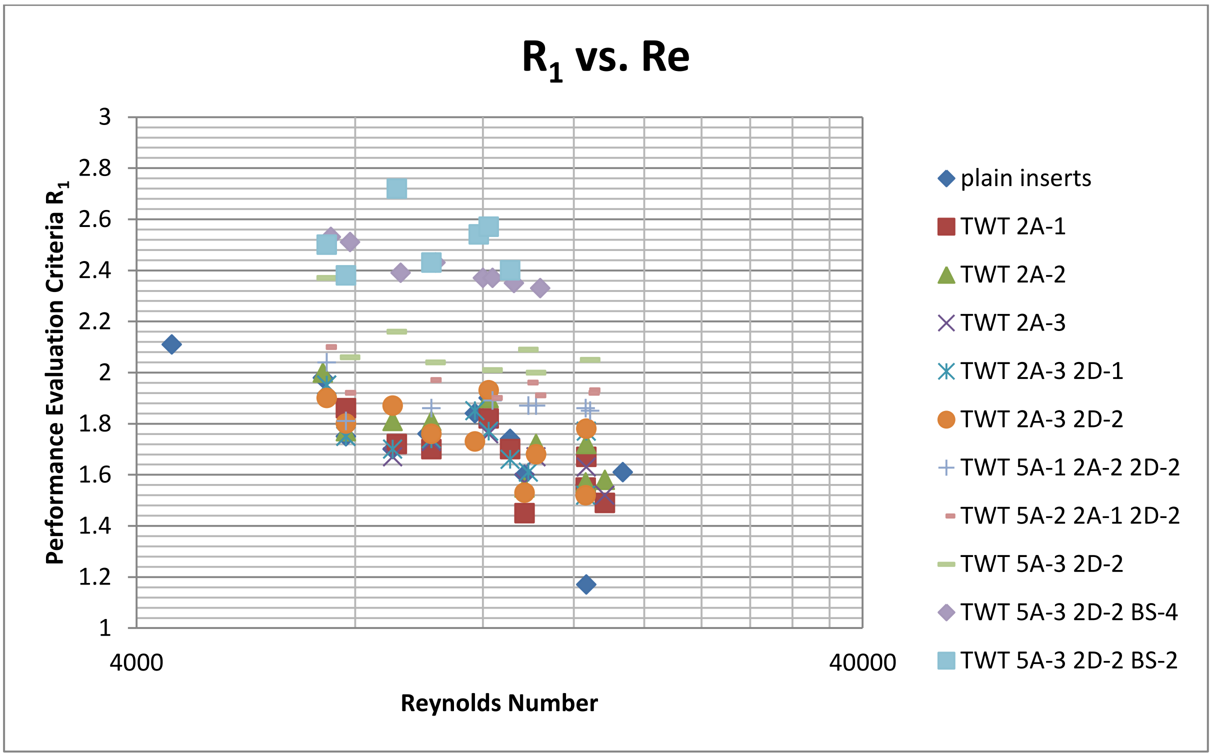


Fig.5.6 Performance Evaluation Criteria R_1 vs. Reynolds Number for plain wavy insert and its 10 different modified designs.

Fig. 5.7 shows the plot of Efficiency factor (η) vs. Reynolds Number. The highest vaule of η is observed for the design TWT 5A-2 2D-2 BS-4 than compared to TWT 5A-3 2D-2 BS-2. This can be because of the reason on increasing the number of baffles or decreasing the baffle spacing, the f_a/f_o ratio increases more than the increase in the performance evaluation factor (R_1).

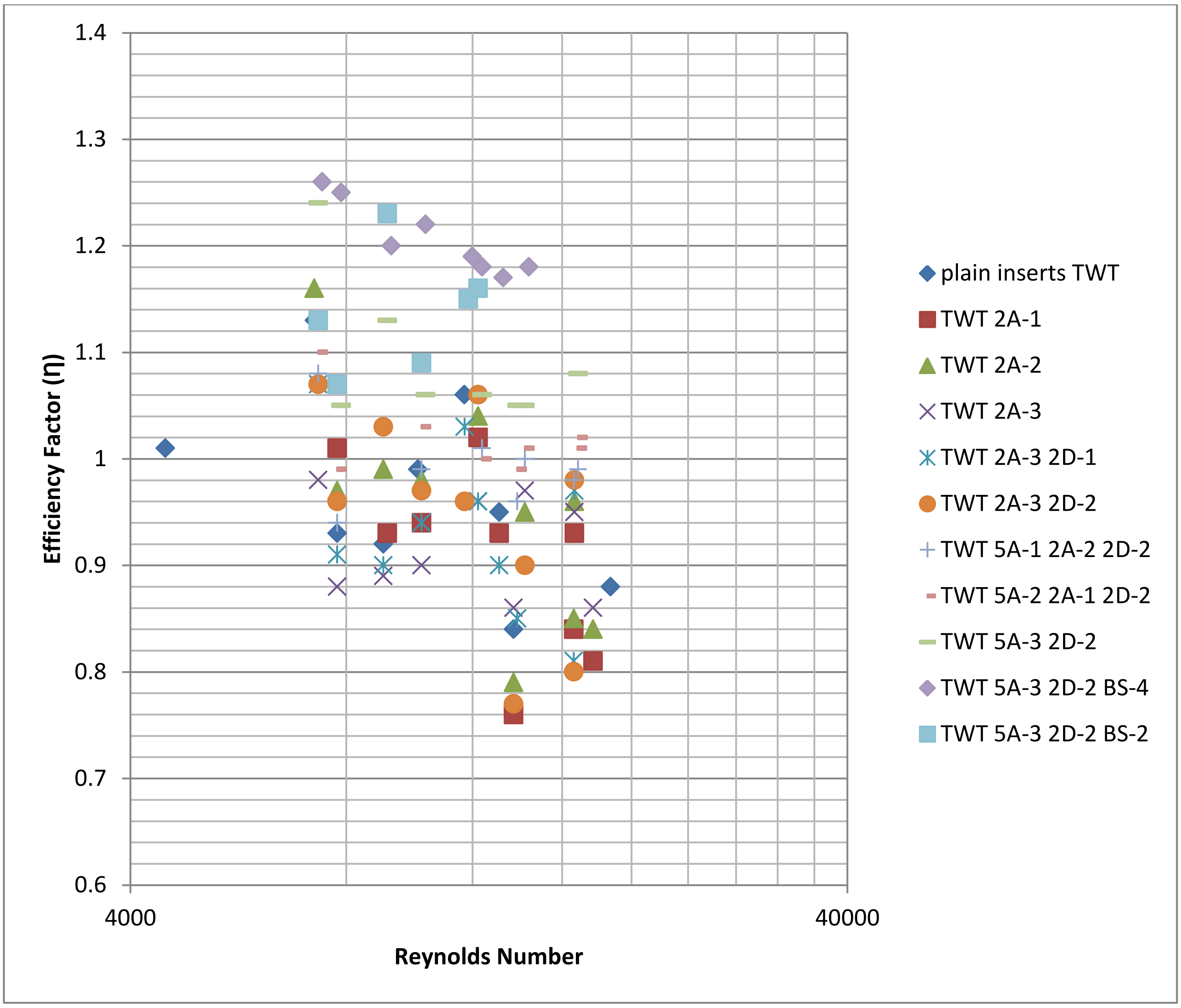


Fig. 5.7 Efficiency factor (η) vs. Reynolds Number for the triangular wavy tape (TWT) and its 10 different modified designs.

Chapter 6

CONCLUSIONS

The range of performance evaluation criteria R1 (based on constant mass flow rate), fa/fo & values of efficiency factor (η) for different inserts used are given below.

. Table 6.1: Range of R_1, f_a/f_o, η for different inserts

Sl.no	Insert	$R_1 = Nu_a/Nu_o$	f_a/f_o	$\eta = R_1/(f_a/f_o)^{1/3}$
1	Triangular Wavy Insert (TWT)	1.53 – 1.98	5.35-7.16	0.84 – 1.13
2	TWT 2A-1	1.45 – 1.86	5.85 – 7.12	0.76 – 1.02
3	TWT 2A-2	1.55 – 2	5.23 – 7.72	0.79 – 1.16
4	TWT 2A-3	1.52 – 1.95	5.68 – 8.13	0.86 – 0.98
5	TWT 2A-3 2D-1	1.52 – 1.95	5.92 – 7.14	0.81 – 1.07
6	TWT 2A-3 2D-2	1.52 – 1.93	5.75 – 7.94	0.77 – 1.07
7	TWT 5A-1 2A-2 2D-2	1.81 – 2.04	6.67 – 7.48	0.94 – 1.08
8	TWT 5A-2 2A-1 2D-2	1.9 – 2.1	6.94 – 7.96	0.99 – 1.10
9	TWT 5A-3 2D-2	2 – 2.37	6.97 – 8.06	1.03 – 1.24
10	TWT 5A-3 2D-2 BS-4	2.33 – 2.53	7.8 – 8.35	1.17 – 1.26
11	TWT 5A-3 2D-2 BS-2	2.4 – 2.72	11.01 – 11.38	1.07 – 1.23

1. From the above table, we can say that with the increase in number and size of the holes, the heat transfer coefficient values and the friction factor increases with the increase in degree of turbulence.
2. From the table, we can observe that the performance evaluation criteria for the design TWT 5A-3 2D-2 BS-2 is the highest indicating that the heat transfer coefficient is about 2.7 times the value of the smooth tube.
3. With the decrease in baffle spacing, both the heat transfer coefficient and the pressure drop increases.
4. The value of efficiency factor (η) is the highest for TWT 5A-3 2D-2 BS-4 because of decrease in value of f_a/f_o ratio is more than decrease in R_1 value, when compared to TWT 5A-3 2D-2 BS-2 case.

Chapter 7

SCOPE FOR FUTURE WORK

Further modification can be done using this study as base. Some of the possibilities are mentioned below:

1. The wavelength of the wave can be varied and their effect on heat transfer coefficient and friction factor can easily be noted down.
2. Pressure drop is a huge loss in this modification. So, studies can be made to minimize the pressure drop.
3. The same experiment can also be done with cooling operations.
4. The experiments can conducted for laminar flow of fluids through the inner tube.

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APPENDIX

A.1 CALIBRATION

A.1.1 ROTAMETER CALIBRATION

Rotameter readings (Kg/hr)	Observation 1			Observation 2			Observation 3			Average (Kg/sec)	Actual Flow (Kg/hr)	%age Diff.
	Wt. (Kg)	Time (sec)	M (Kg/sec)	Wt. (Kg)	Time (sec)	M (Kg/sec)	Wt. (Kg)	Time (sec)	M (Kg/sec)			
300	0.96	16.59	0.0579	0.79	12.86	0.0614	0.8	12.96	0.0617	0.0603	217	27.67
350	0.96	9.97	0.0963	0.81	8.19	0.0989	0.85	8.46	0.1005	0.0986	354	1.14
400	1.18	11.24	0.1050	1.32	11.65	0.1133	1.14	11.4	0.1	0.1061	381	4.75
450	1.11	8.96	0.1239	1.19	9.31	0.1278	1.07	8.76	0.1222	0.1246	448	0.44
500	1.29	9.33	0.1383	1.36	9.21	0.1477	1.15	8.41	0.1367	0.1409	507	1.4
550	1.46	8.9	0.1640	1.49	9.03	0.1650	1.32	8.14	0.1622	0.1637	589	7.09
600	1.44	8.27	0.1741	1.41	8.61	0.1638	1.5	8.56	0.1752	0.1710	615	2.5
650	1.52	8.03	0.1893	1.55	8.35	0.1856	1.56	8.96	0.1741	0.1830	658	1.23
700	1.62	8.18	0.1980	2	9.86	0.2028	1.72	8.8	0.1955	0.1988	715	2.14
750	2.18	11.24	0.1940	2	10.44	0.1916	2.03	10.33	0.1965	0.1940	698	6.93
800	2.21	9.15	0.2415	1.97	8.24	0.2391	1.85	8.15	0.2270	0.2359	849	6.13
850	2.13	8.63	0.2468	2.23	9.05	0.2464	2.2	10.33	0.2130	0.2354	847	0.35
900	2.26	8.9	0.2539	2.05	8	0.2563	2.1	8.72	0.2408	0.2503	901	0.11
950	2.57	9.47	0.2714	2.52	9.47	0.2661	2.59	9.7	0.2670	0.2682	965	1.58
1000	2.46	8.73	0.2818	2.58	8.95	0.2883	2.45	8.28	0.2960	0.2887	1039	3.9
1100	2.27	7.86	0.2888	2.87	10.03	0.2861	2.67	8.44	0.3164	0.2971	1069	2.82
1200	2.39	7.5	0.3187	2.72	7.99	0.3404	2.81	8.11	0.3465	0.3352	1206	0.5

A.1.2 RTD CALIBRATION:

SL. NO.	TEMPERATURE READINGS (^0C)			
	T1	T2	T3	T4
1	14.7	14.5	14.9	15
2	15.2	15	15.3	15.5
3	15.2	15	15.3	15.5
CORRECTION	0	0.2	-0.1	-0.3

A.2 FRICTION FACTOR RESULTS:

A.2.1 STANDARDISATION OF SMOOTH TUBE (f vs. Re)

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	%diff.
0.0986	2.1	26.3	6639	124.22	7.2	7.9	8.86
0.1246	3.7	26.3	8389	218.87	7.9	7.6	-3.95
0.1409	4.4	26.3	9487	260.28	7.4	7.4	0
0.1637	5.5	26.3	11022	325.35	6.8	7.1	4.23
0.171	6	26.3	11513	354.93	6.8	7.1	4.23
0.1988	7.2	26.3	13385	425.91	6	6.9	13.04
0.2359	10.4	26.3	15883	615.2	6.2	6.6	6.06
0.2503	12.2	26.3	16853	721.68	6.5	6.6	1.52
0.2887	15.8	26.3	19438	934.64	6.3	6.4	1.56
0.2971	17.9	26.3	20004	1058.86	6.7	6.3	-6.35
0.3352	20.6	26.3	22569	1218.58	6.1	6.2	1.61
0.3533	23	26.3	23788	1360.55	6.1	6.1	0

A.2.2 FRICTION FACTOR vs. REYNOLDS NUMBER FOR PLAIN WAVY TAPE[10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	fa/fo
0.0603	8.9	26.3	4060	526.47	81.25	8.73	9.31
0.0986	12.7	26.3	6639	751.26	43.36	7.91	5.48
0.1061	18.1	26.3	7144	1070.69	53.35	7.8	6.84
0.1246	22.7	26.3	8389	1342.8	48.54	7.55	6.43
0.1409	25.5	26.3	9487	1508.43	42.64	7.37	5.79
0.1637	30.9	26.3	11022	1827.87	38.26	7.15	5.35
0.171	36.4	26.3	11513	2153.22	41.31	7.09	5.83
0.183	43.6	26.3	12321	2579.13	43.2	6.99	6.18
0.194	56.1	26.3	13062	3318.56	49.48	6.91	7.16
0.2354	69.4	26.3	15850	4105.31	41.57	6.65	6.25
0.2682	87.1	26.3	18058	5152.34	40.19	6.48	6.2

A.2.3 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 2A-1 [10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	fa/fo
0.10609	17	26.5	7143	1005.62	50.1	7.8	6.42
0.12462	22.9	26.5	8391	1354.63	48.9	7.6	6.43
0.14089	26.6	26.5	9486	1573.5	44.5	7.4	6.01
0.17104	36.6	26.5	11516	2165.05	41.5	7.1	5.85
0.18301	44.1	26.5	12322	2608.7	43.7	7	6.24
0.19401	55.7	26.5	13063	3294.89	49.1	6.9	7.12
0.23587	64.9	26.5	15881	3839.11	38.7	6.6	5.86
0.2354	70	26.5	15850	4140.8	41.9	6.6	6.35
0.25034	78.3	26.5	16855	4631.78	41.5	6.6	6.29

A.2.4 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 2A-2 [10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	fa/fo
0.09855	12.1	26.3	6635	715.77	41.3	7.9	5.23
0.10609	16.6	26.3	7143	981.96	49	7.8	6.28
0.12462	22.1	26.3	8391	1307.31	47.2	7.6	6.21
0.14089	28.4	26.3	9486	1679.98	47.5	7.4	6.42
0.17104	38.5	26.3	11516	2277.44	43.7	7.1	6.15
0.19878	50.4	26.3	13384	2981.38	42.3	6.9	6.13
0.19401	60.4	26.3	13063	3572.92	53.3	6.9	7.72
0.23587	65.6	26.3	15881	3880.52	39.1	6.6	5.92
0.2354	71.8	26.3	15850	4247.28	43	6.6	6.52
0.25034	84.1	26.3	16855	4974.88	44.5	6.6	6.74

A.2.5 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 2A-3 [10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	fa/fo
0.09855	18.6	26.4	6635	1100.27	63.6	7.9	8.05
0.10609	21.5	26.4	7143	1271.82	63.4	7.8	8.13
0.12462	24.3	26.4	8391	1437.45	51.9	7.6	6.83
0.14089	31.8	26.4	9486	1881.11	53.2	7.4	7.19
0.17104	38.7	26.4	11516	2289.27	43.9	7.1	6.18
0.19878	42.7	26.4	13384	2525.89	35.9	6.9	5.2
0.19401	47.7	26.4	13063	2821.66	42.1	6.9	6.1
0.23587	56.1	26.4	15881	3318.56	33.5	6.6	5.08
0.25034	70.1	26.4	16855	4146.72	37.1	6.6	5.62

A.2.6 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 2A-3 2D-1 [10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	1000*f_{exp}	1000*f_{theo}	fa/fo
0.09855	14.1	26.4	6635	834.08	48.2	7.9	6.1
0.10609	18.9	26.4	7143	1118.02	55.7	7.8	7.14
0.12462	24.7	26.4	8391	1461.11	52.8	7.6	6.95
0.14089	28.9	26.4	9486	1709.56	48.3	7.4	6.53
0.16374	33.9	26.4	11025	2005.33	42	7.1	5.92
0.17104	39.6	26.4	11516	2342.51	44.9	7.1	6.32
0.18301	45.4	26.4	12322	2685.61	45	7	6.43
0.19401	54.8	26.4	13063	3241.66	48.3	6.9	7
0.23587	67.9	26.4	15881	4016.58	40.5	6.6	6.14
0.2354	74.6	26.4	15850	4412.91	44.7	6.6	6.77

A.2.7 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 2A-3 2D-2 [10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	$1000*f_{exp}$	$1000*f_{theo}$	fa/fo
0.09855	13.3	26.4	6635	786.75	45.4	7.9	5.75
0.10609	17.9	26.4	7143	1058.86	52.8	7.8	6.77
0.12462	21.9	26.4	8391	1295.48	46.8	7.6	6.16
0.14089	26.8	26.4	9486	1585.34	44.8	7.4	6.05
0.16374	33.9	26.4	11025	2005.33	42	7.1	5.92
0.17104	38.8	26.4	11516	2295.19	44	7.1	6.2
0.19878	54.5	26.4	13384	3223.91	45.8	6.9	6.64
0.19401	62.1	26.4	13063	3673.48	54.8	6.9	7.94
0.23587	66.7	26.4	15881	3945.59	39.8	6.6	6.03
0.2354	75.9	26.4	15850	4489.81	45.5	6.6	6.89

A.2.8 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 5A-1 2A-2 2D-2[10]

m (Kg/sec)	ΔH (cm)	T (^{0}C)	Re	ΔP (N/m^2)	$1000*f_{exp}$	$1000*f_{theo}$	fa/fo
0.09855	15.8	26.7	6635	934.64	54	7.9	6.84
0.10609	19	26.7	7143	1123.93	56	7.8	7.18
0.14089	29.8	26.7	9486	1762.8	49.8	7.4	6.73
0.17104	41.8	26.7	11516	2472.65	47.4	7.1	6.68
0.19878	54.9	26.7	13384	3247.57	46.1	6.9	6.68
0.19401	58.5	26.7	13063	3460.53	51.6	6.9	7.48
0.23587	73.8	26.7	15881	4365.59	44	6.6	6.67
0.2354	77.4	26.7	15850	4578.54	46.4	6.6	7.03

A.2.9 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 5A-2 2A-1 2D-2[10]

m (Kg/sec)	**ΔH (cm)**	**T (^{0}C)**	**Re**	**ΔP (N/m^2)**	**1000*f_{exp}**	**1000*f_{theo}**	**fa/fo**
0.09855	16.2	26.7	6635	958.3	55.4	7.9	7.01
0.10609	19.5	26.7	7143	1153.51	57.5	7.8	7.37
0.14089	31.6	26.7	9486	1869.28	52.8	7.4	7.14
0.17104	43.6	26.7	11516	2579.13	49.5	7.1	6.97
0.19878	57.1	26.7	13384	3377.71	48	6.9	6.96
0.19401	62.3	26.7	13063	3685.31	54.9	6.9	7.96
0.23587	76.7	26.7	15881	4537.13	45.8	6.6	6.94
0.2354	77.1	26.7	15850	4560.8	46.2	6.6	7

A.2.10 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 5A-3 2D-2 [10]

m (Kg/sec)	**ΔH (cm)**	**T (^{0}C)**	**Re**	**ΔP (N/m^2)**	**1000*f_{exp}**	**1000*f_{theo}**	**fa/fo**
0.09855	16.6	26.7	6635	981.96	56.7	7.9	7.18
0.10609	20.3	26.7	7143	1200.83	59.9	7.8	7.68
0.12462	25.5	26.7	8391	1508.43	54.5	7.6	7.17
0.14089	32	26.7	9486	1892.94	53.5	7.4	7.23
0.17104	43.6	26.7	11516	2579.13	49.5	7.1	6.97
0.19878	58.1	26.7	13384	3436.86	48.8	6.9	7.07
0.19401	63.1	26.7	13063	3732.64	55.6	6.9	8.06
0.23587	78.1	26.7	15881	4619.95	46.6	6.6	7.06

A.2.11 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 5A-3 2D-2 BS-4 [10]

m (Kg/sec)	**ΔH (cm)**	**T (^{0}C)**	**Re**	**ΔP (N/m^2)**	**1000*f_{exp}**	**1000*f_{theo}**	**fa/fo**
0.09855	19.1	26.9	6713	1129.85	65.3	7.9	8.27
0.10609	21.9	26.9	7227	1295.48	64.6	7.8	8.28
0.12462	28.3	26.9	8489	1674.07	60.5	7.5	8.07
0.14089	35.5	26.9	9598	2099.98	59.4	7.4	8.03
0.16374	46.7	26.9	11154	2762.51	57.8	7.1	8.14
0.17104	52.3	26.9	11652	3093.77	59.3	7.1	8.35
0.18301	58.7	26.9	12467	3472.36	58.2	7	8.31
0.19878	64	26.9	13541	3785.88	53.8	6.9	7.8

A.2.12 FRICTION FACTOR vs. REYNOLDS NUMBER FOR TWT 5A-3 2D-2 BS-2 [10]

m (Kg/sec)	**ΔH (cm)**	**T (^{0}C)**	**Re**	**ΔP (N/m^2)**	**1000*f_{exp}**	**1000*f_{theo}**	**fa/fo**
0.09855	26	26.9	6713	1538.01	88.8	7.9	11.24
0.10609	30.1	26.9	7227	1780.54	88.8	7.8	11.38
0.12462	39.1	26.9	8489	2312.93	83.6	7.5	11.15
0.14089	50	26.9	9598	2957.72	83.6	7.4	11.3
0.16374	64	26.9	11154	3785.88	79.2	7.1	11.15
0.17104	68.9	26.9	11652	4075.73	78.2	7.1	11.01

A.3 HEAT TRANSFER RESULTS:

A.3.1 STANDARDISATION OF SMOOTH TUBE (hi vs. Re)

m (Kg/ sec)	**T_1**	**T_2**	**T_3**	**T_4**	**LMTD**	**U_i**	**Re**	**h_{exp}**	**h_{theo}**	**%diff.**
0.0986	26.3	33.3	52.1	49.5	20.92	841	7137	1720	1515	13.53
0.1246	26.2	32.4	52.6	49.7	21.81	902	8907	1996	1884	5.94
0.1409	26.2	31.8	51.7	49	21.32	900	9950	1986	2092	5.07
0.1637	26.2	32	52.5	49.6	21.92	998	11702	2536	2274	11.52
0.171	26.3	31.5	52.5	49.4	22.03	990	12075	2485	2335	6.42
0.1988	26.3	31.3	52.2	49.1	21.84	1059	14038	2970	2639	12.54
0.2359	26.3	30.6	52.2	48.9	22.1	1089	16658	3219	3038	5.96
0.2503	26.3	30.4	52.3	48.8	22.2	1122	17462	3526	3157	11.69
0.2887	26.3	30.1	52.2	48.6	22.2	1177	20141	4133	3544	16.62
0.2971	26.3	29.7	51.8	48.2	22	1139	20727	3699	3635	1.76
0.3352	26.3	29.5	51.9	48.2	22.15	1182	23385	4195	4009	4.64
0.3533	26.3	29.4	51.9	48.1	22.15	1210	24354	4570	4145	10.25

A.3.2 Heat Transfer Coefficient vs. Re for plain wavy tape

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.0603	26.3	36.6	52.9	50.1	880	4476	1891	896	2.11	1.01
0.0986	26.3	35.1	52.6	49.5	1065	7227	3018	1521	1.98	1.13
0.1061	26.3	34.4	52.5	49.4	1046	7777	2870	1640	1.75	0.93
0.1246	26.3	33.6	52.2	49	1089	9019	3219	1895	1.7	0.92
0.1409	26.3	33	52	48.7	1123	10072	3536	2004	1.76	0.99
0.1637	26.3	32.7	52.3	48.9	1179	11702	4157	2263	1.84	1.06
0.171	26.3	32.5	52.6	49	1201	12224	4444	2345	1.9	1.06
0.183	26.3	32	52	48.5	1191	13082	4311	2484	1.74	0.95
0.194	26.3	31.4	52.4	48.6	1178	13699	4145	2585	1.6	0.84
0.2359	26.3	31.2	50.9	48.3	1124	16658	3546	3028	1.17	---
0.2354	26.3	30.8	51.8	48.1	1215	16623	4642	3028	1.53	0.84
0.2682	26.3	30.4	51.9	48	1259	18711	5358	3336	1.61	0.88

A.3.3 Heat Transfer Coefficient vs. Re for TWT 2A-1

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.1061	26.5	34.9	53.2	50	1067	7776	3034	1634	1.86	1.01
0.1246	26.5	34.1	52.5	49.4	1096	9134	3281	1911	1.72	0.93
0.1409	26.5	33.4	52.8	49.5	1113	10198	3438	2018	1.7	0.94
0.1710	26.5	32.6	52.5	49	1187	12227	4259	2342	1.82	1.02
0.1830	26.5	32.2	52.7	49.1	1183	13083	4208	2478	1.7	0.93
0.1940	26.5	31.5	52.3	48.7	1143	13700	3742	2582	1.45	0.76
0.2359	26.5	31.4	52.4	48.8	1241	16656	5046	3021	1.67	0.93
0.2354	26.5	31	52.3	48.5	1217	16623	4672	3023	1.55	0.84
0.2503	26.5	30.9	52.6	48.8	1222	17678	4746	3176	1.49	0.81

A.3.4 Heat Transfer Coefficient vs. Re for TWT 2A-2

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.3	35.3	52.9	49.8	1067	7223	3034	1519	2	1.16
0.1061	26.3	34.5	53	49.8	1049	7776	2893	1639	1.77	0.97
0.1246	26.3	33.9	52.7	49.4	1112	9020	3429	1893	1.81	0.99
0.1409	26.3	33.4	52.6	49.3	1135	10198	3657	2019	1.81	0.98
0.1710	26.3	32.5	52.6	49	1202	12227	4458	2346	1.9	1.04
0.1988	26.3	31.9	52.7	49.1	1209	14210	4556	2656	1.72	0.95
0.1940	26.3	31.5	52.5	48.8	1167	13700	4012	2584	1.55	0.79
0.2359	26.3	31.2	52.7	48.9	1250	16656	5199	3028	1.72	0.96
0.2354	26.3	30.9	52.3	48.5	1222	16623	4746	3028	1.57	0.85
0.2503	26.3	30.7	52.4	48.5	1240	17678	5030	3183	1.58	0.84

A.3.5 Heat Transfer Coefficient vs. Re for TWT 2A-3

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.4	35.5	52.9	49.9	1062	7316	2994	1536	1.95	0.98
0.1061	26.4	34.7	53	49.9	1044	7776	2855	1636	1.75	0.88
0.1246	26.4	33.9	52.8	49.6	1083	9020	3167	1892	1.67	0.89
0.1409	26.4	33.1	52.5	49.1	1120	10198	3506	2021	1.73	0.9
0.1710	26.4	32.5	52.9	49.3	1177	12227	4133	2346	1.76	0.96
0.1988	26.4	31.9	52.7	49.1	1200	14210	4431	2654	1.67	0.97
0.1940	26.4	31.5	52.3	48.6	1170	13700	4048	2584	1.57	0.86
0.2359	26.4	31.1	52.5	48.7	1234	16656	4933	3028	1.63	0.95
0.2503	26.4	30.7	52.1	48.3	1227	17678	4823	3180	1.52	0.86

A.3.6 Heat Transfer Coefficient vs. Re for TWT 2A-3 2D-1

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.4	35.6	52.7	49.8	1062	7316	2994	1535	1.95	1.07
0.1061	26.4	34.7	52.6	49.6	1045	7776	2863	1636	1.75	0.91
0.1246	26.4	34	52.9	49.7	1089	9020	3219	1890	1.7	0.9
0.1409	26.4	33.4	52.4	49.2	1121	10198	3516	2018	1.74	0.94
0.1637	26.4	32.9	52.6	49.2	1180	11705	4170	2260	1.85	1.03
0.1710	26.4	32.7	53	49.5	1179	12227	4157	2342	1.77	0.96
0.1830	26.4	32.2	52.3	48.9	1175	13083	4108	2480	1.66	0.9
0.1940	26.4	31.7	52.6	48.9	1181	13869	4182	2605	1.61	0.85
0.2359	26.4	31.3	52	48.4	1259	16656	5358	3023	1.77	0.97
0.2354	26.4	31.1	52.2	48.6	1212	16623	4599	3023	1.52	0.81

A.3.7 Heat Transfer Coefficient vs. Re for TWT 2A-3 2D-2

m (Kg/ sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	η= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.4	35.6	52.9	50	1052	7316	2916	1535	1.9	1.07
0.1061	26.4	34.5	52.4	49.3	1056	7776	2947	1639	1.8	0.96
0.1246	26.4	34	52.6	49.3	1123	9020	3536	1890	1.87	1.03
0.1409	26.4	33.4	52.7	49.4	1124	10198	3546	2018	1.76	0.97
0.1637	26.4	32.9	52.3	49.1	1158	11705	3908	2260	1.73	0.96
0.1710	26.4	32.6	52.6	49	1207	12227	4528	2344	1.93	1.06
0.1988	26.4	31.8	52.4	48.8	1202	14210	4458	2656	1.68	0.9
0.1940	26.4	31.5	52.1	48.5	1162	13700	3953	2584	1.53	0.77
0.2359	26.4	31.3	52.3	48.6	1261	16656	5394	3023	1.78	0.98
0.2354	26.4	31.1	52.2	48.6	1212	16623	4599	3023	1.52	0.8

A.3.8 Heat Transfer Coefficient vs. Re for TWT 5A-1 2A-2 2D-2

m (Kg/ sec)	**T_1**	**T_2**	**T_3**	**T_4**	**U_i**	**Re**	**h_a**	**h_o**	**R1= h_a/h_o**	**η= $R_1/(f_a/f_o)^{1/3}$**
0.0986	26.7	35.9	53.1	50.1	1077	7316	3117	1531	2.04	1.08
0.1061	26.7	34.8	52.3	49.3	1058	7776	2963	1634	1.81	0.94
0.1409	26.7	33.5	52.6	49.2	1143	10198	3742	2014	1.86	0.99
0.1710	26.7	32.7	52.2	48.7	1202	12380	4458	2362	1.89	1.01
0.1988	26.7	32.1	52.8	49	1235	14210	4949	2646	1.87	1.00
0.1940	26.7	31.9	52.8	48.8	1230	13869	4869	2599	1.87	0.96
0.2359	26.7	31.5	52.1	48.4	1273	16861	5621	3046	1.85	0.99
0.2354	26.7	31.3	52.5	48.5	1272	16623	5602	3014	1.86	0.98

A.3.9 Heat Transfer Coefficient vs. Re for TWT 5A-2 2A-1 2D-2

m (Kg/ sec)	**T_1**	**T_2**	**T_3**	**T_4**	**U_i**	**Re**	**h_a**	**h_o**	**R1= h_a/h_o**	**η= $R_1/(f_a/f_o)^{1/3}$**
0.0986	26.7	36	52.7	49.8	1088	7316	3211	1530	2.1	1.10
0.1061	26.7	34.9	52.8	49.6	1080	7776	3142	1633	1.92	0.99
0.1409	26.7	33.7	53	49.5	1162	10198	3953	2011	1.97	1.03
0.1710	26.7	32.8	52.4	48.9	1204	12380	4486	2360	1.9	1.00
0.1988	26.7	32.2	52.6	48.9	1242	14210	5063	2646	1.91	1.01
0.1940	26.7	32	52.8	48.8	1244	13869	5096	2597	1.96	0.99
0.2359	26.7	31.6	52.8	48.9	1286	16861	5884	3043	1.93	1.02
0.2354	26.7	31.4	52.9	48.8	1284	16828	5842	3041	1.92	1.01

A.3.10 Heat Transfer Coefficient vs. Re for TWT 5A-3 2D-2

m (Kg/sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	Ƞ= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.7	36.1	52.8	49.7	1132	7316	3626	1530	2.37	1.24
0.1061	26.7	35.2	52.8	49.6	1109	7876	3401	1651	2.06	1.05
0.1246	26.7	34.8	52.9	49.6	1176	9134	4120	1904	2.16	1.13
0.1409	26.7	34	53	49.6	1177	10327	4133	2028	2.04	1.06
0.1710	26.7	33.1	53.1	49.5	1221	12380	4731	2356	2.01	1.06
0.1988	26.7	32.4	53.2	49.4	1255	14210	5286	2641	2	1.05
0.1940	26.7	32.2	52.3	48.5	1263	13869	5431	2595	2.09	1.05
0.2359	26.7	31.7	52.5	48.7	1302	16861	6234	3041	2.05	1.08

A.3.11 Heat Transfer Coefficient vs. Re for TWT 5A-3 2D-2 BS-4

m (Kg/sec)	T_1	T_2	T_3	T_4	U_i	Re	h_a	h_o	R1= h_a/h_o	Ƞ= $R_1/(f_a/f_o)^{1/3}$
0.0986	26.9	36.3	52.9	49.7	1158	7411	3908	1546	2.53	1.26
0.1061	26.9	35.8	53.2	49.8	1176	7876	4120	1644	2.51	1.25
0.1246	26.9	35	52.5	49.2	1212	9251	4599	1922	2.39	1.2
0.1409	26.9	34.6	52.7	49.4	1233	10327	4917	2020	2.43	1.22
0.1637	26.9	33.8	52.9	49.4	1262	12001	5413	2287	2.37	1.19
0.1710	26.9	33.4	52.7	49.1	1270	12380	5563	2350	2.37	1.18
0.1830	26.9	32.8	52.5	48.7	1284	13246	5842	2489	2.35	1.17
0.1988	26.9	32.5	52.7	48.8	1301	14388	6211	2663	2.33	1.18

A.3.12 Heat Transfer Coefficient vs. Re for TWT 5A-3 2D-2 BS-2

m (Kg/ sec)	**T_1**	**T_2**	**T_3**	**T_4**	**U_i**	**Re**	**h_a**	**h_o**	**R1= h_a/h_o**	**η= $R_1/(f_a/f_o)^{1/3}$**
0.0986	26.3	35.9	52.4	49.3	1151	7316	3829	1534	2.5	1.13
0.1061	26.3	35	52.6	49.2	1156	7776	3885	1635	2.38	1.07
0.1246	26.3	34.4	52.9	49.1	1250	9134	5199	1910	2.72	1.23
0.1409	26.3	33.9	52.9	49.3	1232	10198	4901	2014	2.43	1.09
0.1637	26.3	33.3	52.2	48.7	1281	11851	5780	2279	2.54	1.15
0.1710	26.3	32.8	52.7	48.8	1292	12227	6011	2342	2.57	1.16
0.1830	26.3	32.5	52.5	48.7	1289	13083	5947	2476	2.4	---